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The design of spur and helical gearing

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THE DESIGN OF SPUR AND HELICAL GEARING

A Thesis submitted in partial fulfilment of the requirements for the award of the degree of

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by

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This is to certify that this work has not been submitted for a degree to any other University or Institution.

R.J. Davey

DEDICATION

This Thesis is dedicated to my late mother, Annie Lorraine, whose untiring efforts made this publication possible, and who impressed upon me from a young age that the fear of the Lord was the beginning of wisdom.

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SYNOPSIS

This Thesis is the result of a fifteen year involvement in both the theoretical and practical aspects of spur and helical gear design.

Over this period of time, the evolution of the computer, has rendered its use in gear design to be almost mandatory. However, the greatest problem faced in writing the necessary software, has been the analysis of the Lewis parabola inscribed within the gear tooth profile.

The tenet of this Thesis, is the presentation of an original analytical method for the calculation of the height and width of the Lewis parabola. This theory has been adopted, both in an analytical and graphical format, by the Australian Standards Association Gear Committee ME/11, and has been distributed as AS 2938-1987, being a supplement to AGMA 218.01 Dec 1982.

An adaptation of the theory, has resulted in the production of software for the rating and/or design of gears to AGMA 218.01. The package is designed to be used by a person unfamiliar with gear design, whilst at the same time, the program has sufficient flexibility to satisfy the needs of an experienced gear designer.

To complement the theory, and as part of the ongoing program of gear research being conducted at the University of Wollongong, a gear testing rig was designed and manufactured. Although four gear sets do not provide sufficient evidence from which to draw categorical conclusions, the trends were towards a verification of the theory, for modified addendums, in combination with minimum specific sliding velocities.

ACKNOWLEDGEMENTS

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Thanks are also due to Messrs Nevison, Robinson and Wotherspoon of BHP Steel International, Slab & Plate Products Division, Port Kembla, for their assistance and consideration during the preparation of this Thesis, and in particular to Ms Cheryl Bryant and Mrs Valmai Rose for the care and patience they displayed in the typing and proofreading of this manuscript.

LIST OF SYMBOLS

BHN = Brinell Hardness Number

B_N = backlash applied in the normal plane, mm

b = operating dedendum, mm

C = operating centre distance, mm

C_a = application factor for pitting resistance

C_c = curvature factor at pitch line

C_e = mesh alignment correction factor

C_f = surface condition factor

C_H = hardness ratio factor

C_h = helical factor

C_I = involute clearance coefficient

C_L = life factor for pitting resistance

C_m = load distribution factor for pitting resistance

C_{ma} = mesh alignment factor

C_{mc} = lead correction factor

C_{mf} = face load distribution factor

C_{mt} = transverse load distribution factor

C_o = overlap coefficient

C_p = elastic coefficient, $(MPa)^{1/2}$

C_{pf} = pinion proportion factor

C_{pm} = pinion proportion modifier

C_R = reliability factor for pitting resistance

C_s = size factor for pitting resistance

C_T = temperature factor for pitting resistance

C_U = bottom clearance coefficient

C_v = dynamic factor for pitting resistance

C_W = tip (top land) width coefficient

C_x	=	contact height factor
C_{xh}	=	contact height factor for LCR gears
C_ψ	=	helical overlap factor
D	=	operating pitch diameter of wheel, mm
d	=	operating pitch diameter of pinion, mm
E	=	modulus of elasticity, MPa
F	=	net facewidth of narrowest member, mm
FRAC	=	fractional portion
f_P	=	surface finish of pinion, rms
H	=	Dolan and Broghamer factor H
H_B	=	Brinell hardness
HRC	=	Rockwell "C" Hardness Number
h	=	radial distance from the critical section to the intersection of load line with the centreline of the tooth, mm
h_a	=	addendum of the counterpart rack, mm
h_b	=	dedendum of the counterpart rack, mm
h_e	=	height of the stress parabola derived from the virtual tooth profile, mm
$[h_e]$	=	reference height of the stress parabola
h_{eo}	=	height of the stress parabola, mm, for a value of $\lambda = \lambda_o$
I	=	pitting resistance geometry factor
[I]	=	reference pitting resistance geometry factor
INT	=	integer portion
inv	=	involute function: inv $(\phi_c) = \tan(\phi_c) - \phi_c$
J	=	bending strength geometry factor
[J]	=	reference bending strength geometry factor
K_A	=	application factor
K_a	=	application factor for bending strength
K_f	=	stress correction factor

$K_{H\alpha}$	=	transverse load factor for contact stress
$K_{H\beta}$	=	longitudinal load factor for contact stress
K_L	=	life factor for bending strength
K_m	=	load distribution factor for bending strength
K_R	=	reliability factor for bending strength
K_s	=	size factor for bending strength
K_T	=	temperature factor for bending strength
K_v	=	dynamic factor for bending strength **or** dynamic factor
K_ψ	=	helix angle factor
k	=	total cutter profile shift, mm
K_1	=	geometrical factor of the basic rack, mm
K_2	=	geometrical constant of the basic rack, mm
K_3	=	constant of integration
K_4	=	angle of obliquity factor
K_5	=	first approximation of obliquity factor, rad
K_6	=	first iteration of obliquity factor, rad
K_7	=	second iteration of obliquity factor, rad
K_8	=	angle of rotation of gear blank, rad
K_9	=	geometrical constant of the stress parabola
L	=	Dolan and Broghamer factor L
L_{min}	=	minimum length of lines of contact, mm
M	=	Dolan and Broghamer factor M
M_I	=	reference pitting resistance geometry factor multiplier
M_J	=	reference bending strength geometry factor multiplier
M_{he}	=	reference parabola height multiplier
M_{te}	=	reference parabola width multiplier
m_F	=	face contact ratio

m_G	=	gear ratio (always ≥ 1.0)
m_N	=	load sharing ratio
m_n	=	normal metric module, mm
m_p	=	transverse contact ratio
N	=	number of teeth or number of load cycles
n	=	rotational speed, rpm, or limiting number of lines of contact
P_{ac}	=	allowable transmitted power for pitting resistance, kW
P_{at}	=	allowable transmitted power for bending strength, kW
P_b	=	BS 436: Part 1 strength rating, kW
P_c	=	BS 436: Part 1 wear rating, kW
P_σ	=	surface durability rating, kW
p	=	normal pitch, mm
p_a	=	axial pitch, mm
p_b	=	transverse base pitch, mm
p_{bn}	=	normal base pitch, mm
p_N	=	normal base pitch, mm
Q_v	=	transmission accuracy level number
R	=	operating pitch radius of wheel, mm
R_F	=	radius to top of the tooth fillet, mm
R_o	=	tip radius of wheel, mm
R_{om}	=	maximum allowable tip radius of wheel, mm
r	=	operating pitch radius of pinion, mm
r_T	=	edge radius of the cutting tool, mm
r_f	=	minimum root fillet radius, mm
r_o	=	tip radius of pinion, mm
r_{om}	=	maximum allowable tip radius of pinion, mm
r_r	=	root radius of the pinion, mm

S = bearing span, mm, **or** slide to roll ratio

S_b = bending stress factor, MPa

S_c = surface stress factor, MPa

S_{Hmin} = safety factor for contact stress

S_1 = pinion offset, mm

s_{ac} = allowable contact stress number, MPa

s_{at} = allowable bending stress number, MPa

t = tooth thickness at the critical section, mm

t_e = width of the stress parabola derived from the virtual tooth profile, mm

$[t_e]$ = reference width of the stress parabola

t_{eo} = width of the stress parabola, mm, for a value of $\lambda = \lambda_o$

t_o = normal chordal tooth thickness at the top land, mm

t_{ot} = transverse arc tooth thickness at the top land, mm

t_{st} = transverse reference arc tooth thickness, mm

u = radial distance from tooth layout, mm

V_r = rolling velocity, m/s

V_s = sliding velocity, m/s

v_t = pitch line velocity at operating pitch diameter, m/s

W_d = incremental dynamic load, N

W_n = normal tooth reaction, N

W_t = transmitted tangential load, N

X_b = strength speed factor

X_c = wear speed factor

x = addendum modification coefficient with no backlash

Y = tooth form factor **or** strength factor

Y_F = tooth form factor for bending stress

Y_S = stress correction factor for bending strength

Y_β = helix angle factor for bending stress

Y_ϵ = contact ratio factor

Z = length of line of action in the transverse plane, mm, or zone factor

Z_a = length of the recess path, mm

Z_b = length of the approach path, mm

Z_c = length from pitch point to stress point, mm

Z_{ch} = distance from pitch point to stress point for LCR gears, mm

Z_E = elasticity factor for contact stress, MPa

Z_H = zone factor for Hertzian at pitch point for contact stress

Z_L = lubricant factor for contact stress

Z_N = life factor for contact stress

Z_R = roughness factor for contact stress

Z_W = work hardening factor for contact stress

Z_X = size factor for contact stress

Z_{ne} = equivalent length of line of action for LCR gears, mm

Z_v = speed factor for contact stress

Z_β = helix angle factor for contact stress

Z_ϵ = contact ratio factor for contact stress

β = angle between tooth tip and base circle, degrees

ΔR_o = radial tooth truncation applied to the wheel, mm

Δr_o = radial tooth truncation applied to the pinion, mm

ϵ = slide to roll ratio

ξ = accuracy constant

λ = angle between tooth centreline and stress tangent, degrees

λ_o = an arbitrary choice of λ, degrees

λ_1 = the revised value of λ, degrees

μ = Poisson's Ratio

σ_{Hlim}	=	endurance limit, MPa
ϕ_L	=	load angle at HPSTC, degrees
ϕ_{LN}	=	load angle at tooth tip, degrees
ϕ_c	=	normal profile angle of equivalent rack cutter, degrees
ϕ_n	=	normal operating pressure angle, degrees
ϕ_o	=	tip transverse pressure angle, degrees
ϕ_s	=	transverse pressure angle at standard diameter, degrees
ϕ_t	=	operating transverse pressure angle, degrees
ψ	=	helix angle at operating pitch diameter, degrees
ψ_b	=	base helix angle, degrees
ψ_o	=	tip helix angle, degrees
ψ_s	=	helix angle at standard pitch diameter, degrees
Ω_h	=	projected transverse half tip angle, rad
ω	=	load inclination angle, degrees, **or** angular velocity, rad/s

Subscripts

G	=	wheel
P	=	pinion
b	=	base diameter
e	=	equivalent (virtual)
o	=	outside diameter
s	=	reference

INTRODUCTION

The American Gear Manufacturers Association Standard, AGMA 218.01 Dec. 1982, gives the power rating formulae for spur and helical gearing as,

$$P_{ac} = \frac{n_P\, F}{1.91 \times 10^7} \; \frac{I\, C_v}{C_s\, C_m\, C_f\, C_a} \left[\frac{d\, s_{ac}\, C_L\, C_H}{C_p\, C_T\, C_R} \right]^2$$

where P_{ac} = allowable transmitted power for pitting resistance, kW,

and

$$P_{at} = \frac{n_P\, d\, K_v}{1.91 \times 10^7\, K_a} \; F\, m \; \frac{J}{K_s\, K_m} \; \frac{s_{at}\, K_L}{K_R\, K_T}$$

where P_{at} = allowable transmitted power for bending strength, kW.

Thus the power capacities for strength and wear of both the pinion and wheel can be determined, the allowable power capacity for the combination being the lowest of the four values. These equations tend to become "checking" equations for gears already in service. In their present form they do not lend themselves to the design of new gears.

The work described in this Thesis is based on the author's fifteen year continuous involvement in gear design in heavy industry. One of the basic aims of this Thesis is to take most of the "guesswork" out of the procedures for the design of spur and helical gears by the development of computer software. This aim has been satisfied by the production of software for the rating and/or design of spur and helical gears to AGMA 218.01 which is the basis of the current Australian Standard, AS 2938 June 1987.

SECTION 1

THE LAMBDA METHOD - A METHOD FOR THE DETERMINATION OF THE HEIGHT AND WIDTH OF THE LEWIS PARABOLA

1.1 Introduction

Since 1893, when W Lewis (1) published the concept of inscribing a parabola within a gear tooth to analyse the bending stress, various graphical and analytical methods have been proposed to determine the height and width of the Lewis equal strength parabola.

In addressing the problem of determining the height and width of the parabola, two schools of thought exist; namely, that of a draftsman's layout such as detailed in AGMA 226.01 (2) or an analytical method.

For a gear set where teeth numbers, addendum modifications and cutter details are known, the state of the art regarding gear tooth bending strength analysis is such that the precision of a geometry factor taken from a draftsman's layout can compare favourably with the precision with which many of the variables in the power rating equation are determined.

However, in looking at the broader problem of designing a set of gears for a known application, many permutations of teeth numbers, modules, pressure angles, helix angles, addendum modifications, centre distances and materials are possible. Many of these combinations would require different tooth profile layouts for the subsequent calculation of the power rating. In this situation, a computer program in combination with a suitable algorithm, would ideally lend itself to the determination of the most suitable combination, if the calculation of the strength geometry factory were analytical.

Appreciating that several authors, including Errichello (3) have presented an analytical method for the determination of the strength geometry factor, this treatise presents an analytical method which yields the AGMA 218.01 (4) strength geometry factor to an order of accuracy commensurate with a draftsman's layout without the need for iteration. However, if a more accurate answer is required, one iteration will yield a strength geometry factor correct to three decimal places.

This rapid convergence has been achieved by a judicious choice of the dependent variable in the iteration. This choice has also overcome the problem of non-convergence that has been known to arise in similar analytical methods for very large teeth numbers and/or severely modified teeth. This non-convergence has been a result of the Lewis parabola being "mathematically inverted", ie, opening upwards rather than downwards.

Should a national gear rating code other than AGMA 218.01 be used, the relevant equations for the calculation of the height and width of the Lewis equal strength parabola can be isolated and then incorporated into the pertinent strength geometry factor of the national code being utilised.

The method is readily adaptable to computerisation. A listing of the computer program developed for calculating strength geometry factors may be found in Appendix A.

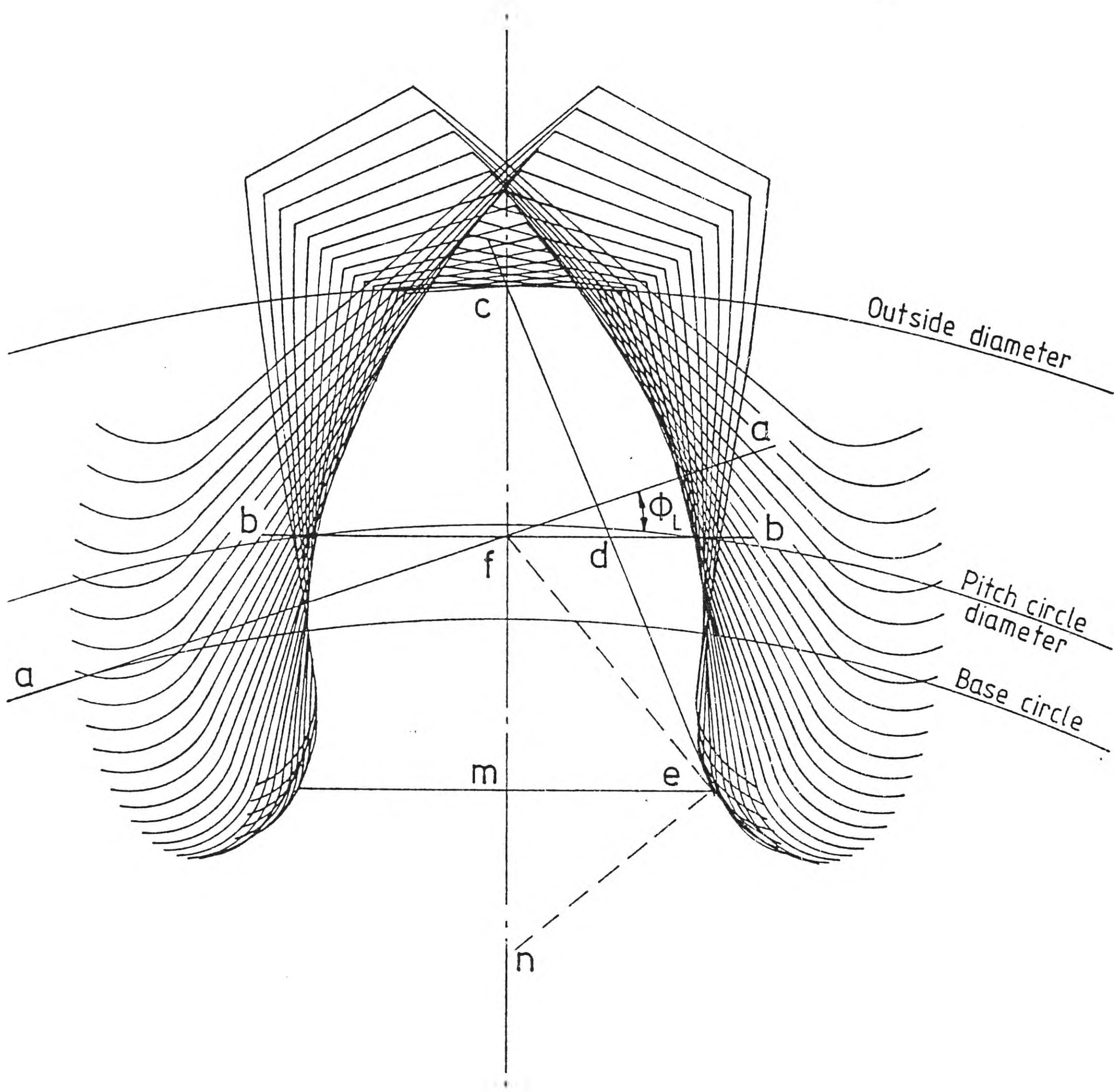


FIGURE 1.1 - COMPUTER SIMULATION OF TOOTH LAYOUT

1.2 Equivalence of Parabola and Equal Line Segments Approaches

Prior to the determination of the strength geometry factor, J, the form factor, Y, is calculated from the dimensions of the Lewis parabola.

As its name suggests, the form factor is a function of the tooth geometry and therefore, information regarding, the tooth shape is necessary for its calculation. In AGMA 218.01, Y is calculated by the following formula:

$$Y = \frac{K_{\psi} \cos(\phi_n)}{\cos(\phi_L)\ (1.5/\ u\ C_h\ -\ \tan(\phi_L)/t)} \tag{1.1}$$

(assuming unit normal module)

The AGMA method for the calculation of Y requires that the tooth dimensions t and u, be obtained from a layout of the actual (normal) tooth profile, the tooth profile having been obtained via a manual generation process. Figure 1.1 shows a computer simulation of this generation process, whilst Figure 1.2 shows the resulting tooth profile.

Referring to Figure 1.2, the tooth dimensions t and u are derived from a construction whereby the line ce is drawn such that cd = de and is tangent to the fillet portion of the tooth. The line bb (which contains point d) is drawn perpendicular to the tooth centre-line through point f. The load line aa is found by drawing a line through point f, tangential to the base circle.

In the method that has been developed, a slightly different, but mathematically identical pair of dimensions are used to represent the tooth shape. These dimensions, (h and t on Figure 1.3), are found from the height and width of an inverted parabola which has its apex at f and is tangential to the fillet portion of the tooth.

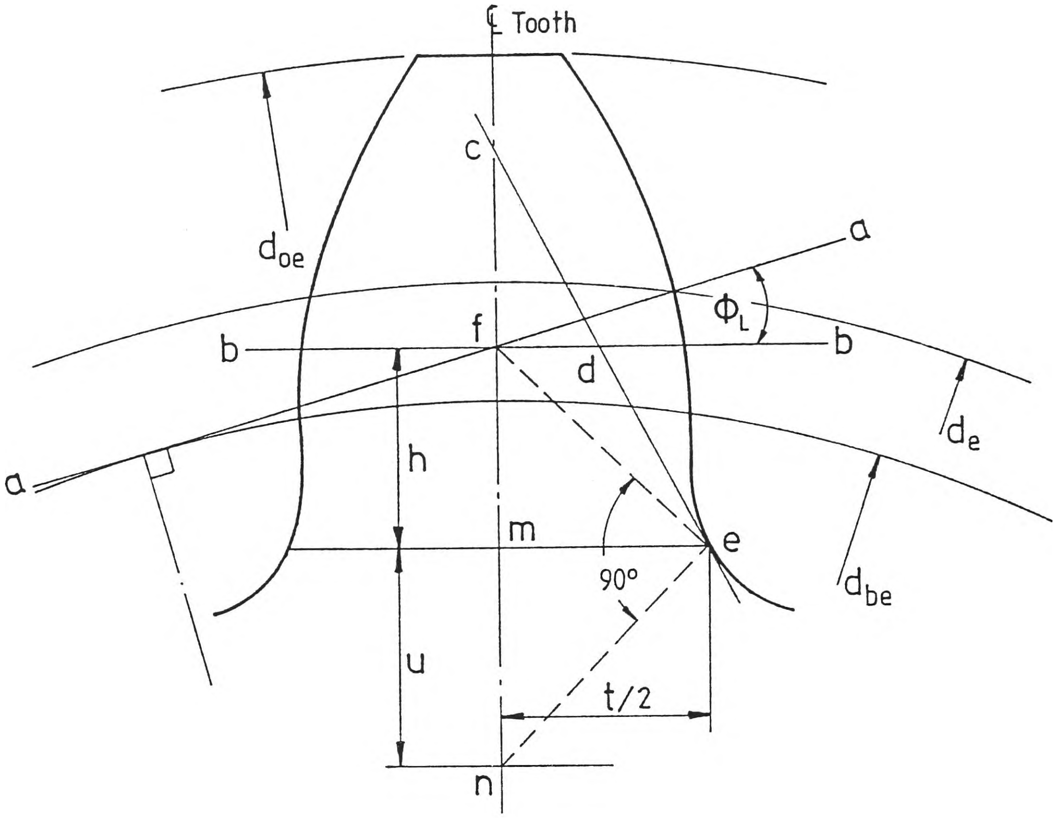


FIGURE 1.2 - AGMA 218.01 TOOTH DIMENSIONS

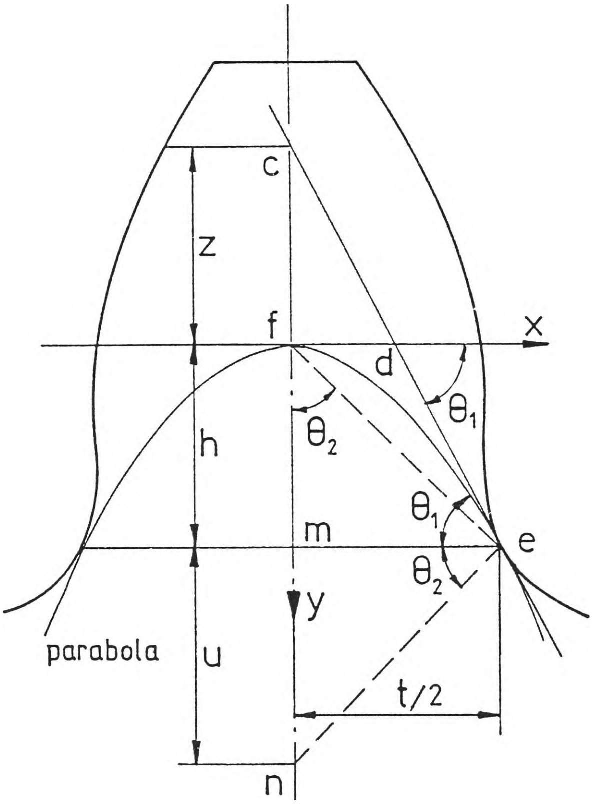


FIGURE 1.3 - LAMBDA METHOD TOOTH DIMENSIONS

The following proof will show the existence of a direct relationship between the two pairs of tooth dimensions.

The general form of the equation of a parabola which has its apex at the origin is,

$$y = ax^2$$

therefore,

$$dy/dx = 2ax$$

With reference to the parabola on the x - y system of Figure 1.3,

$$h = a\,(t/2)^2$$

$$\therefore\ a = 4h/t^2$$

At point e, then,

$$dy/dx = 2(4h/t^2(t/2))$$

ie, $$dy/dx = 4h/t$$

$$\therefore\ \theta_1 = \mathrm{atan}(4h/t)$$

Now, the angle cem is equal to θ_1 (alternate angles).

In triangle cme,

$$\tan(\theta_1) = (z + h)/(t/2) = dy/dx = 4h/t$$

$$\therefore\ z = h$$

But, triangles cfd and cme are similar,

$$\therefore\ cf/fm = cd/de = 1$$

from which cd = de

ie, the equal line segment approach and parabolic approach are equivalent methods for finding point e.

Now triangles fme and emn are similar.

$$\therefore\ fm/me = em/mn$$

from which $$mn = em^2/fm$$

Substituting mn = u, em = t/2 and fm = h results in,

$$u = t^2/4h \tag{1.6}$$

By combining equations (1.1) and (1.6), the following alternate expression for the tooth form factor can be derived.

$$Y = \frac{K_\psi\ t^2 \cos(\phi_n)}{\cos(\phi_L)\ [6\ h/C_h - t\ \tan(\phi_L)]}$$

By using the virtual number of teeth principle, the above equation can be rewritten as

$$Y = \frac{K_\psi\ t_e^2 \cos(\phi_n)}{\cos(\phi_L)\ [6\ h_e/C_h - t_e\ \tan(\phi_L)]} \tag{1.7}$$

where the dimensions t_e and h_e are derived from the virtual tooth profile.

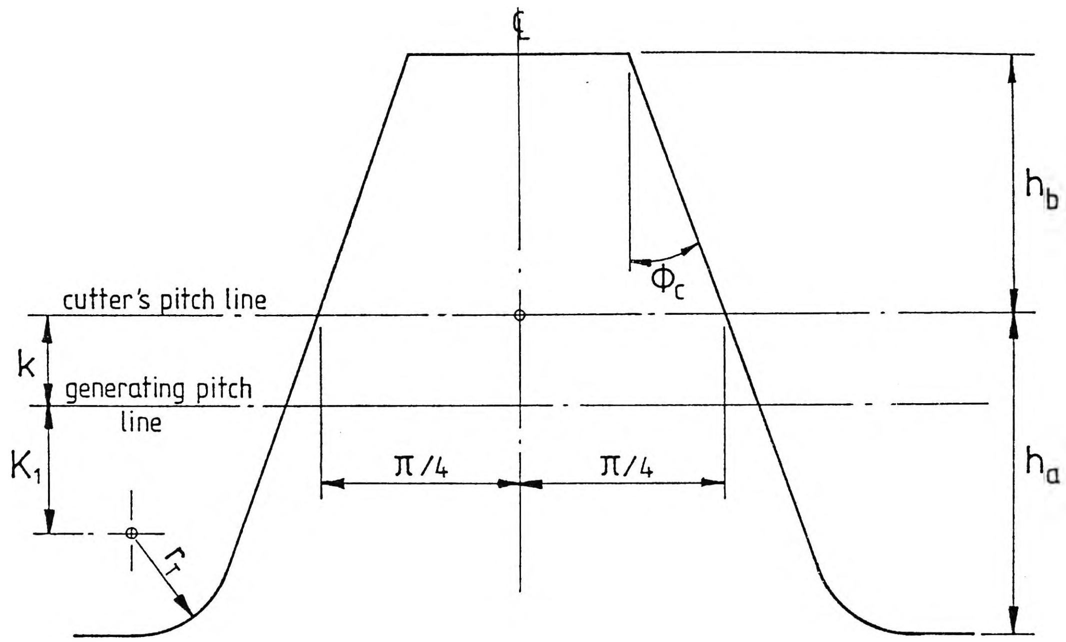


FIGURE 1.4 - RACK CUTTER CONSTANTS

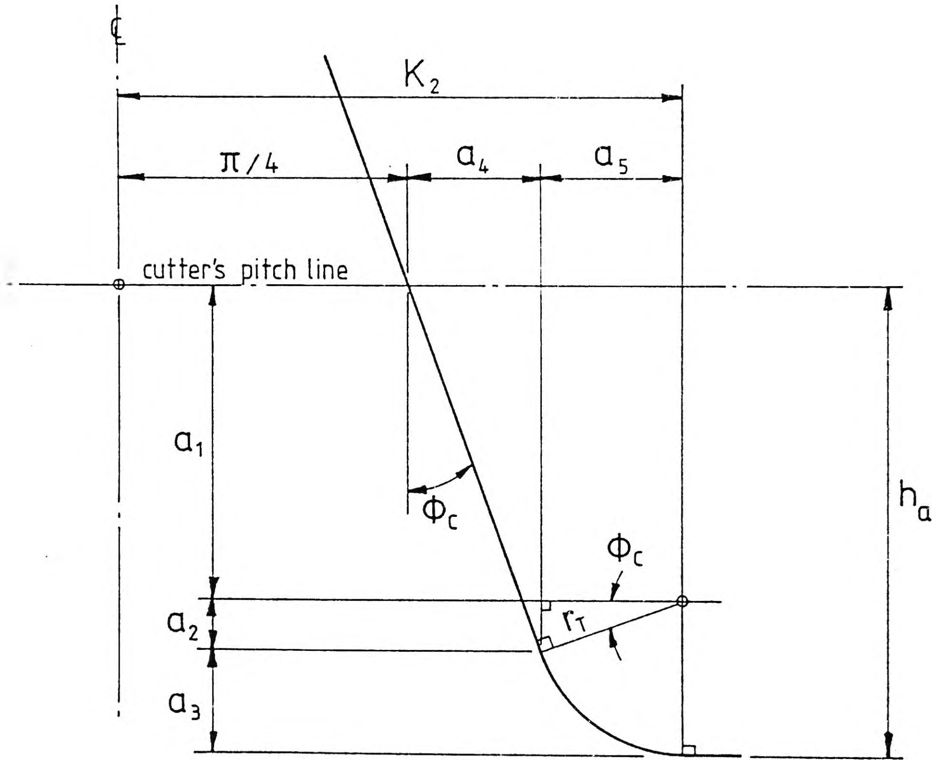


FIGURE 1.5 - TIP RADIUS LOCATION

1.3 Rack Cutter Constants

Before embarking on the derivation of the Lambda Method, it is advantageous to define certain rack dimensions which are constant in the mathematical analysis.

Consider Figures 1.4 and 1.5. By inspection,

$$K_1 = h_a - r_T - k^* \tag{1.8}$$

$$h_a = a_1 + a_2 + a_3$$

$$a_2 = r_T \sin(\phi_c)$$

$$a_3 = r_T (1 - \sin(\phi_c))$$

$$a_1 + a_2 = h_a - r_T (1 - \sin(\phi_c))$$

$$a_4 = (a_1 + a_2) \tan(\phi_c)$$

$$a_4 = (h_a - r_T (1 - \sin(\phi_c)))\tan(\phi_c)$$

$$a_5 = r_T \cos(\phi_c)$$

$$K_2 = 0.25\pi + a_4 + a_5$$

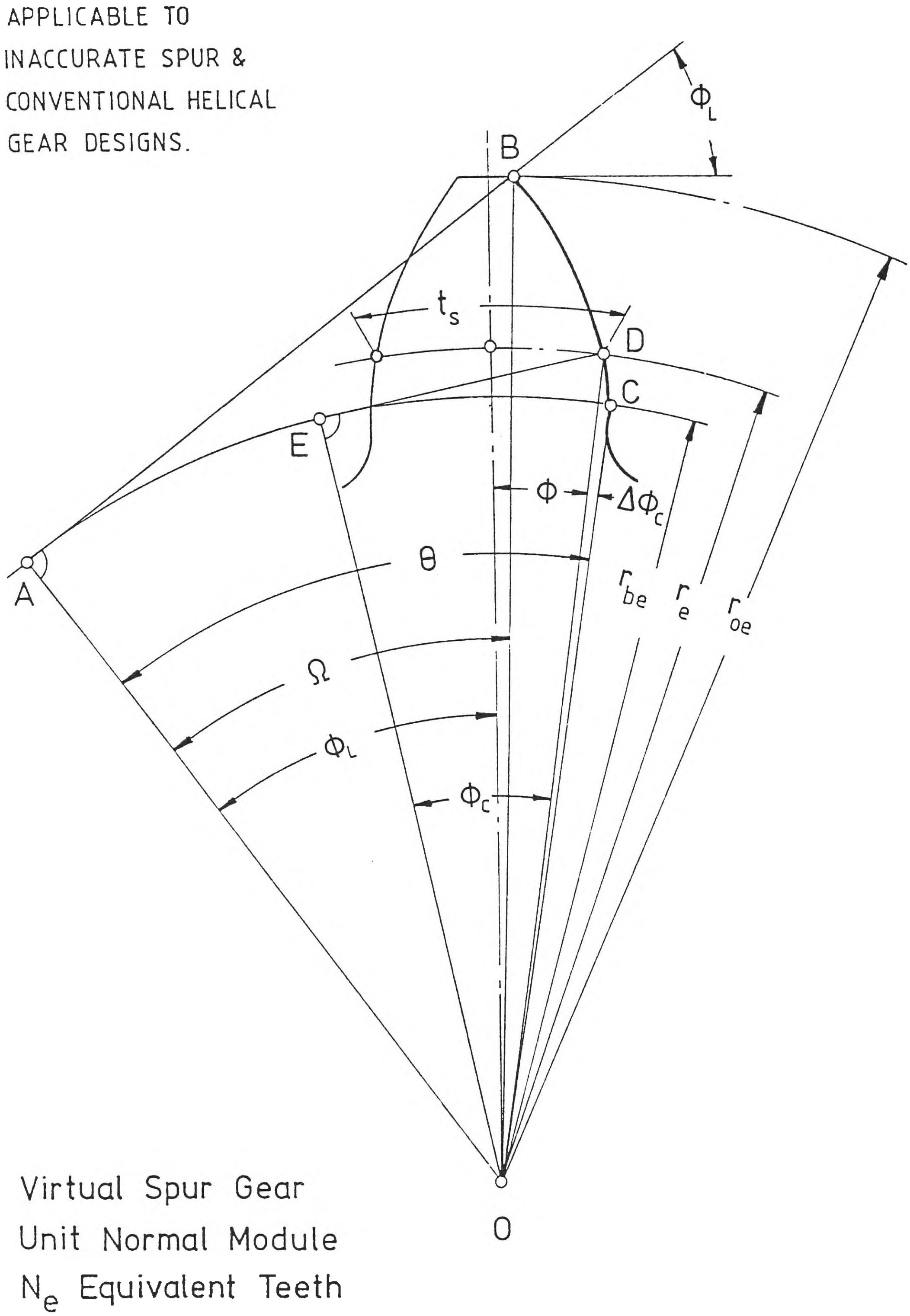
$$K_2 = 0.25\pi + (h_a - r_T (1 - \sin(\phi_c)))\tan(\phi_c) + r_T \cos(\phi_c)$$

$$K_2 = 0.25\pi + h_a \tan(\phi_c) + r_T (1 - \sin(\phi_c))/\cos(\phi_c) \tag{1.9}$$

* Note: Here k represents the total cutter profile shift, ie, inclusive of backlash, B_N.

$$k = x - 0.5\ B_N/\tan(\phi_c)$$

$$K_1 = h_a - r_T - x + 0.5\ B_N/\tan(\phi_c) \tag{1.8a}$$



$$\boxed{\Delta\Phi_c = \mathrm{inv}(\Phi_c)}$$

FIGURE 1·6 - LOAD ANGLE AT TOOTH TIP

1.4 Load Angle Calculation

Prior to the derivation of the Lambda Method, the equations for the calculation of ϕ_L, the load angle, are established.

It can be seen from Figures 1.2 and 1.3, that the load angle, ϕ_L, has a direct effect on the magnitudes of h and t. In AGMA 218.01 two basic philosophies are used to calculate ϕ_L.

These are:

1. Tooth tip loading, and
2. Highest point of single tooth contact loading (HPSTC).

Tooth tip loading is assumed for inaccurate spur gears and conventional helical gears and HPSTC loading is assumed for accurate spur gears and low contact ratio (LCR) helical gears. In AGMA 218.01 the procedure for finding ϕ_L is a graphical one, so it became necessary to investigate the use of analytical techniques for determining ϕ_L in the gear design package program.

Referring to Figure 1.6, and recognising that an involute may be represented by the locus of any fixed point on a taut string which is being unwound from a stationary circle, then

$$ED = \text{arc } EC$$

$$ED = r_{be} \tan(\phi_c)$$

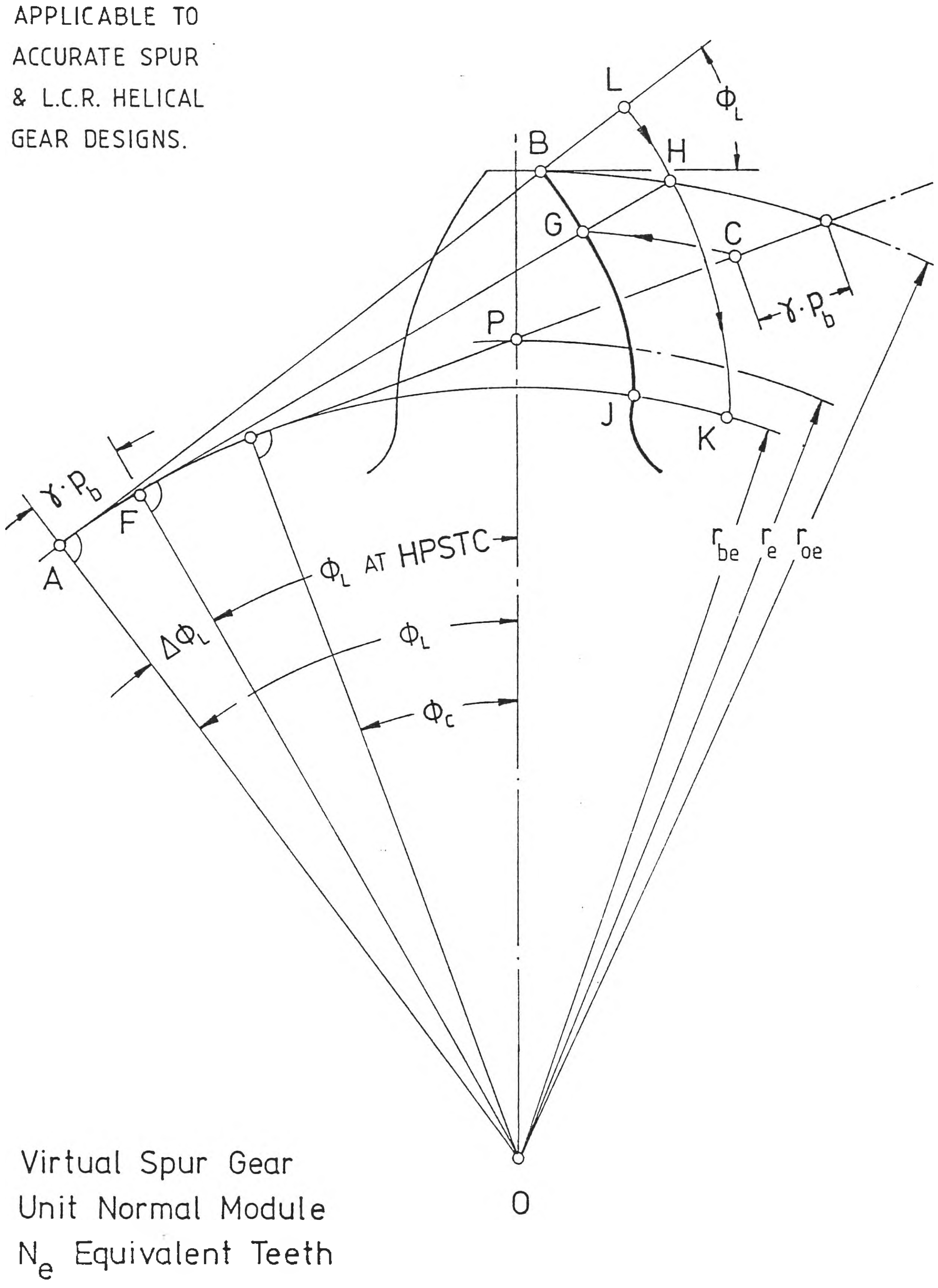
$$\text{arc } EC = r_{be}(\phi_c + \Delta\phi_c)$$

$$\therefore r_{be} \tan(\phi_c) = r_{be}(\phi_c + \Delta\phi_c)$$

$$\text{ie,} \quad \Delta\phi_c = \tan(\phi_c) - \phi_c$$

As the quantity $(\tan(\phi_c) - \phi_c)$, figures prominently in many calculations concerned with the geometry of involute gear teeth, it is universally known as the involute function of the angle ϕ_c, ie,

$$\text{inv}(\phi_c) = \tan(\phi_c) - \phi_c \qquad (1.10)$$



$$\boxed{\gamma \cdot p_b = Z - p_b \text{ AT HPSTC}}$$

FIGURE 1.7 - LOAD ANGLE AT HPSTC

Now, $\cos \Omega = r_{be}/r_{oe}$

$$= 0.5\ N_e \cos(\phi_c)/r_{oe}$$

$$\therefore \quad \Omega = \cos^{-1}(0.5\ N_e \cos(\phi_c)/r_{oe}) \tag{1.11}$$

From the definition of an involute,

$$AB = \text{arc } AC$$

ie,

$$r_{be} \tan(\Omega) = r_{be}\ \theta$$

$$\therefore\ \theta = \tan(\Omega) \tag{1.12}$$

Further, the arc tooth thickness at the reference pitch circle diameter (PCD) is,

$$t_s = 0.5\ \pi + 2k \tan(\phi_c)$$

By inspection of Figure 1.6

$$\phi_L = \theta - \phi - \Delta\phi_c$$

$$= \tan(\Omega) - t_s/N_e - \text{inv}(\phi_c)$$

ie, $$\phi_L = \tan(\Omega) - (0.5\ \pi + 2k \tan(\phi_c))/N_e - \text{inv}(\phi_c) \tag{1.13}$$

where Ω is defined by equation (1.11).

The derivation of the load angle at the HPSTC is shown in Figure 1.7. When the point of contact C, on the path of contact, is rotated about point 0, to intersect the tooth profile, the angle ϕ_L, calculated for tip contact, is reduced by $\Delta\phi_L$.
From the definition of an involute, if the line FGH is wound around the base circle, then the arc distance JK will be equal to $\gamma \cdot p_b$. Further, if the line ABL is wound around the base circle, then BL will coincide with JK. From this observation the arc length AF equals the arc length JK.

Now $\quad \Delta\phi_L$ = Length of arc/radius

$$= \gamma . p_b/(0.5\ N_e \cos(\phi_c))$$

By definition, the base pitch is

$$p_b = \pi \cos(\phi_c)$$

$$\therefore\ \Delta\phi_L = 2\pi\ \gamma/N_e \tag{1.14}$$

In particular, the load point chosen by AGMA for accurate spur gears is the HPSTC, which occurs when γ is equal to the transverse contact ratio less one.

By definition, the transverse contact ratio is,

$$\begin{aligned} m_p &= \text{length of the contact path/base pitch} \\ &= Z/p_b \\ \therefore \gamma &= Z/p_b - 1 \end{aligned} \tag{1.15}$$

Substituting equation (1.15) into equation (1.14) gives

$$\Delta\phi_L = \frac{2\pi}{N_e}\left[\frac{Z}{\pi \cos(\phi_c)} - 1\right]$$

$$= \frac{2\pi}{N_e}\left[\frac{Z - \pi \cos(\phi_c)}{\pi \cos(\phi_c)}\right]$$

ie, $$\Delta\phi_L = 2\ (Z - \pi \cos(\phi_c))/(N_e \cos(\phi_c)) \tag{1.16}$$

The complete sequence for the calculation of the load angles is given in Section 1.9.

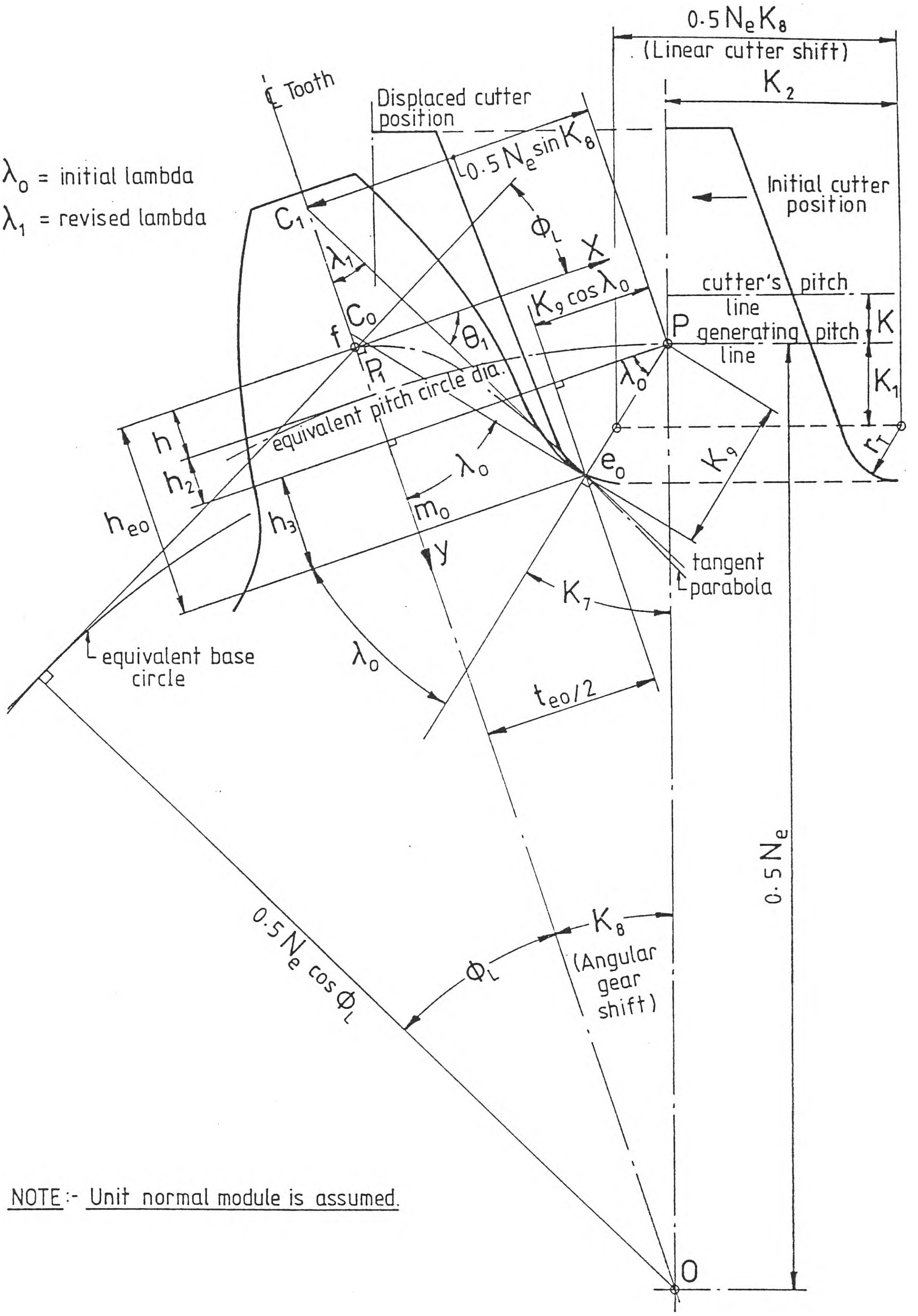


FIGURE 1.8 - DERIVATION OF THE LAMBDA METHOD

1.5 Tooth Form Factor Calculation (Lambda Method)

The derivation of the Lambda Method will be illustrated by means of Figure 1.8 which shows a rack meshing with a gear. From the initial cutter position, assume the rack is moved to the left while the gear rotates counter-clockwise through an angle K_8. Assuming unit normal module, then, the linear displacement of the rack is 0.5 N_e K_8 (N_e = virtual number of teeth). In this position, a line drawn through point P and mutually normal to the tooth root fillet and the cutter tip radius (point "e_o" on Figure 1.8) forms an angle λ_o with a line drawn perpendicular to the centreline of the tooth.

By inspection,

$$K_7 = \pi/2 - \lambda_o - K_8 \tag{1.17}$$

$$K_8 = \frac{K_1 \tan(K_7) + K_2}{0.5 \ N_e} \tag{1.18}$$

$$K_9 = K_1/\cos(K_7) + r_T \tag{1.19}$$

Letting $K_3 = 0.5\ N_e/K_1$ and combining equations (1.17) and (1.18) results in

$$\tan(K_7)/K_3 + K_7 = \pi/2 - \lambda_o - 2K_2/N_e \tag{1.20}$$

The above equation may be solved for angle K_7 by Newton's Method, ie,

iterate:

$$x_{i+1} = x_i - f(x_i)/f'(x_i) \tag{1.21}$$

$$\text{until} \left| x_{i+1} - x_i \right| < \xi$$

where ξ = accuracy constant

The first "guess" for the Newton's Method routine may be obtained from equation (1.21), by letting

$$K_4 = \pi/2 - \lambda_o - 2K_2/N_e$$

and assuming

$$\tan(K_7) = K_7$$

Then $K_7/K_3 + K_7 = K_4$

from which the first "guess" is,

$$K_7 = K_3\ K_4/(K_3 + 1)$$

Now $$f(x) = \tan(K_7)/K_3 + K_7 - K_4 \tag{1.22}$$

$$\therefore \quad f'(x) = \sec^2(K_7)/K_3 + 1 \tag{1.23}$$

Therefore, the Newton's Method iteration scheme to solve equation (1.20) is

$$K_{7i+1} = K_{7i} - \frac{\tan(K_{7i}) + K_3(K_{7i} - K_4)}{\sec^2(K_{7i}) + K_3} \tag{1.24}$$

until $\left| K_{7i+1} - K_{7i} \right| < \xi$

The distances fm_o and e_om_o on Figure 1.8 can now be determined.

$$fm_o = h_{eo} = h_1 + h_2 + h_3$$

$$h_1 = 0.5\ N_e\ (\cos(\phi_c)/\cos(\phi_L) - 1)$$

$$h_2 = 0.5\ N_e\ (1 - \cos(K_8))$$

$$h_3 = (K_1\ \sec(K_7) + r_T)\ \sin(\lambda_o) = K_9\ \sin(\lambda_o)$$

$$h_{eo} = 0.5\ N_e(\cos(\phi_c)/\cos(\phi_L) - \cos(K_8)) + K_9\sin(\lambda_o) \qquad (1.25)$$

$$e_o m_o = t_{eo}/2 = 0.5\ N_e\ \sin(K_8) - K_9\ \cos(\lambda_o)$$

$$t_{eo} = N_e\ \sin(K_8) - 2\ K_9\ \cos(\lambda_o) \qquad (1.26)$$

The above pair of equations can be used to find the distances $e_o m_o$ and fm_o for a particular value of λ_o. The problem still remains, however, in finding the value of λ_o which allows a parabola to be fitted such that its apex is at f while being tangential to the root fillet radius - this being the premise by which the critically stressed section of the tooth is found. The highly transcendental nature of equations (1.25) and (1.26), however, prohibits an exact solution. In response to this problem, an iterative procedure based on the method of direct iteration was formulated and will now be presented.

Assume that for a particular set of gearing parameters h_a, ϕ_c, r_T, N_e and ϕ_L an arbitary choice of λ, ie, $\lambda = \lambda_o$ has resulted in $h_e = h_{eo}$ and $t_e = t_{eo}$ and some point $e = e_o$ on the tooth fillet. If now a parabola with its apex at f and passing through e_o is drawn, then its equation with respect to the x-y system on Figure 1.8 is

$$y = (4\ h_{eo}/t_{eo}^2)x^2 \qquad (1.27)$$

Now the angle θ_1 is equal to the inverse tangent of the derivative of equation (1.27) with respect to x, evaluated at $x = t_{eo}/2$.

$$\left.\frac{dy}{dx}\right|_{e_o} = \left.\frac{8\ h_{eo}.x}{t_{eo}^2}\right|_{x\ =\ t_{eo}/2}$$

$$\text{ie,} \quad \left.\frac{dy}{dx}\right|_{e_o} = \frac{4\ h_{eo}}{t_{eo}}$$

from which $\quad \theta_1 = \mathrm{atan}(4h_{eo}/t_{eo}) \quad$ (1.28)

A line drawn tangent to the parabola (ie, at angle θ_1 to the x axis) and intersecting the centre-line of the tooth at C_1 produces the revised value of lambda, λ_1, ie,

$$\lambda_1 = 90^o - \mathrm{atan}(4h_{eo}/t_{eo})$$

Re-arranging the above equation gives

$$\mathrm{atan}(4h_{eo}/t_{eo}) = 90^o - \lambda_1$$

from which

$$4h_{eo}/t_{eo} = \tan(90^o - \lambda_1)$$

$$= \sin(90^o - \lambda_1)/\cos(90^o - \lambda_1)$$

$$= \cos(\lambda_1)/\sin(\lambda_1)$$

ie, $\quad \lambda_1 = \mathrm{atan}(0.25\ t_{eo}/h_{eo}) \quad$ (1.29)

In general, the above equations may be represented by the following function symbolisation:

$h_{eo} = h_e(\lambda_o)$ (equation (1.25))

$t_{eo} = t_e(\lambda_o)$ (equation (1.26))

$$\lambda_1 = \mathrm{atan}\left[\frac{0.25\ t_e(\lambda_o)}{h_e(\lambda_o)}\right] \quad \text{(equation (1.29))}$$

Letting,

$$F(\lambda_o) = \mathrm{atan}(0.25\ t_e(\lambda_o)/h_e(\lambda_o)) \tag{1.30}$$

$$\therefore\ \lambda_1 = F(\lambda_o)$$

The general form of the iteration procedure can now be formalised as

iterate:

$$\lambda_{i+1} = F(\lambda_i) \tag{1.31}$$

until $\left| \lambda_{i+1} - \lambda_i \right| < \xi$

where ξ = accuracy constant

1.6 Proof of Convergence of Lambda Method

For the iteration scheme depicted by equation (1.31) to be complete, the convergence ability of the procedure must be demonstrated. To this end, a proof was developed which allows the convergence ability to be assessed.

In mathematical parlance, the iteration scheme described by equation (1.31) is referred to as "Direct Iteration", ie,

$$x_{i+1} = F(x_i) \tag{1.32}$$

Now suppose the true solution is x^*, where

$$x^* = F(x^*) \tag{1.33}$$

Subtracting equation (1.33) from equation (1.32) results in

$$x_{i+1} - x^* = F(x_i) - F(x^*)$$

Now the condition for convergence is,

$$\left| x_{i+1} - x^* \right| < \left| x_i - x^* \right|$$

$$\left| F(x_i) - F(x^*) \right| < \left| x_i - x^* \right|$$

ie, $$\left| \frac{F(x_i) - F(x^*)}{x_i - x^*} \right| < 1$$

assuming $x^* = x_i + \Delta x$

$$\left| \frac{F(x_i + \Delta x) - F(x_i)}{\Delta x} \right| < 1$$

ie, $$\left| F'(x_i) \right| < 1 \text{ as } \Delta x \to 0 \tag{1.34}$$

where F' is the derivative of F with respect to x

Equation (1.34) is the general form for the proof of convergence of a direct iteration scheme. This criterion for convergence will now be applied to the iteration scheme described by equation (1.31).

Recalling equation (1.30) and letting $\lambda = \lambda_o$ gives

$$F(\lambda) = \mathrm{atan}(0.25\ t_e(\lambda)/h_e(\lambda))$$

Substituting for t_e and h_e in the above by using equations (1.25) and (1.26) gives

$$F(\lambda) = \mathrm{atan}\left[\frac{0.25\ [N_e \sin(K_8) - 2\ K_9 \cos(\lambda)]}{0.5\ N_e\ [\cos(\phi_c)/\cos(\phi_L) - \cos(K_8)] + K_9 \sin(\lambda)}\right] \quad (1.35)$$

For convenience in the differentiation let

$$u = 0.25\ [N_e \sin(K_8) - 2K_9\cos(\lambda)] \quad (1.36)$$

$$v = 0.5N_e\ [\cos(\phi_c)/\cos(\phi_L) - \cos(K_8)]+ K_9\sin(\lambda) \quad (1.37)$$

and $w = u/v$

ie, $F(\lambda) = \mathrm{atan}(w)$

Now $$\frac{dF(\lambda)}{dw} = \frac{1}{1 + w^2}$$

Hence $$\frac{dF(\lambda)}{d\lambda} = \frac{1}{1 + w^2} \cdot \frac{dw}{d\lambda}$$

By the quotient rule,

$$\frac{dw}{d\lambda} = \frac{d(u/v)}{d\lambda} = \frac{vdu/d\lambda - udv/d\lambda}{v^2}$$

$$\therefore \quad \frac{dF(\lambda)}{d\lambda} = \frac{vdu/d\lambda - udv/d\lambda}{v^2 + u^2} \qquad (1.38)$$

Recalling equation (1.36),

$$u = 0.25\ [N_e \sin(K_8) - 2\ K_9 \cos(\lambda)]$$

where $K_8 = f(\lambda)$ and $K_9 = f(\lambda)$ and in general when

$$y = f[u(x)] \qquad\qquad y' = f'(u).u'(x)$$

$$y = u(x)v(x) \qquad\qquad y' = u'v + uv'$$

then,

$$\frac{du}{d\lambda} = 0.25\ [N_e \cos(K_8)\ \frac{dK_8}{d\lambda} + 2\ K_9 \sin(\lambda) - 2 \cos(\lambda)\ \frac{dK_9}{d\lambda}] \qquad (1.39)$$

Similarly, recalling equation (1.37),

$$v = 0.5\ N_e\ [\cos(\phi_c)/\cos(\phi_L) - \cos(K_8)] + K_9 \sin(\lambda)$$

For a particular gear tooth, ϕ_L is a constant for helical gears (ie, tip loading), but will vary for spur gears, depending on the number of teeth in the mating gear (ie, HPSTC loading). However, prior to the calculation of the parabola dimensions, ϕ_L is predetermined for a spur gear tooth, being independent of λ. Hence, ϕ_L can be considered as a constant when differentiating equation (1.37).

$$\frac{dv}{d\lambda} = 0.5\ N_e \sin(K_8)\ \frac{dK_8}{d\lambda} + K_9 \cos(\lambda) - \sin(\lambda)\ \frac{dK_9}{d\lambda} \qquad (1.40)$$

Consider λ:

From equation (1.20) and an arbitrary choice of $\lambda = \lambda_o$,

$$\lambda = 0.5\pi - 2\ K_2/N_e - K_7 - \tan(K_7)/K_3 \qquad (1.41)$$

ie, $\qquad \lambda = f(N_e,\ K_2,\ K_3,\ K_7)$

where $\quad K_3 = 0.5\ N_e/K_1$

$$K_1 = h_a - r_T - k \qquad (1.8)$$

$$K_2 = 0.25\pi + h_a\ \tan(\phi_c) + r_T\ (1 - \sin(\phi_c))/\cos(\phi_c) \qquad (1.9)$$

$$\therefore \lambda = f(h_a,\ r_T,\ \phi_c,\ k,\ N_e,\ K_7)$$

But $f(h_a,\ r_T,\ \phi_c) = f(\text{cutter})$

For an ISO 53 cutter, which has been chosen in this analysis because of its acceptance by the Standards Association of Australia as the standard cutter in AS 2938 (5),f(cutter) = constant.

ie, $\qquad \lambda = f(k,\ N_e,\ K_7) \qquad (1.42)$

where k represents the total cutter profile shift, ie, inclusive of backlash, B_N.

Consider K_7:

From equation (1.20),

$$y = \tan(K_7)/K_3 + K_7 = 0.5\pi - \lambda - 2K_2/N_e$$

$$\frac{dK_7}{d\lambda} = \frac{dK_7}{dy} \cdot \frac{dy}{d\lambda}$$

where $\quad \dfrac{dy}{dK_7} = \dfrac{\sec^2(K_7)}{K_3} + 1 = \dfrac{\sec^2(K_7) + K_3}{K_3}$

$$\frac{dy}{d\lambda} = -1$$

$$\therefore \frac{dK_7}{d\lambda} = \frac{-K_3}{\sec^2(K_7) + K_3} \tag{1.43}$$

ie, $$\frac{dK_7}{d\lambda} = f(h_a, r_T, k, N_e, K_7)$$

$$\therefore \quad \frac{dK_7}{d\lambda} = f(k, N_e, K_7) \text{ as } f(h_a, r_t) = f(\text{cutter}) = \text{a constant} \tag{1.44}$$

<u>Consider K_8</u>:

From equation (1.17) and an arbitrary choice of $\lambda = \lambda_o$,

$$K_8 = 0.5\pi - \lambda - K_7$$

From equation (1.42),

$$K_8 = f(k, N_e, K_7) \tag{1.45}$$

ie, $$\frac{dK_8}{d\lambda} = -1 - \frac{dK_7}{d\lambda} \tag{1.46}$$

From equation (1.44),

$$\frac{dK_8}{d\lambda} = f(k, N_e, K_7) \tag{1.47}$$

<u>Consider K_9</u>:

From equation (1.19),

$$K_9 = K_1 \sec(K_7) + r_T$$

From equation (1.8), $K_1 = f(h_a, r_T, k)$

$$K_9 = f(k, K_7) \text{ as } f(h_a, r_T) = f(\text{cutter}) = \text{a constant} \tag{1.48}$$

$$\therefore \quad \frac{dK_9}{d\lambda} = \frac{dK_9}{dK_7} \cdot \frac{dK_7}{d\lambda}$$

ie, $$\frac{dK_9}{d\lambda} = K_1 \tan(K_7) \sec(K_7) \frac{dK_7}{d\lambda} \tag{1.49}$$

From equations (1.8) and (1.44)

$$\frac{dK_9}{d\lambda} = f(k, N_e, K_7) \text{ as } K_1 = f(h_a, r_T, k) \tag{1.50}$$

Recalling equation (1.36),

$$u = 0.25 \ [N_e \sin(K_8) - 2 K_9 \cos(\lambda)]$$

From equations (1.45), (1.48) and (1.42)

$$u = f(k, N_e, K_7) \tag{1.51}$$

Similarly, recalling equation (1.37),

$$v = 0.5 \ N_e \ [\cos(\phi_c)/\cos(\phi_L) - \cos(K_8)] + K_9 \sin(\lambda)$$

From equations (1.45), (1.48) and (1.42),

$$v = f(\phi_L, k, N_e, K_7) \tag{1.52}$$

Recalling equation (1.39),

$$\frac{du}{d\lambda} = 0.25 \ [N_e \cos(K_8) \frac{dK_8}{d\lambda} + 2 K_9 \sin(\lambda) - 2 \cos(\lambda) \frac{dK_9}{d\lambda}]$$

From equations (1.45), (1.47), (1.48), (1.42) and (1.50),

$$\frac{du}{d\lambda} = f(k, N_e, K_7) \tag{1.53}$$

Similarly, recalling equation (1.40),

$$\frac{dv}{d\lambda} = 0.5 \ N_e \sin(K_8) \frac{dK_8}{d\lambda} + K_9 \cos(\lambda) + \sin(\lambda) \frac{dK_9}{d\lambda}$$

$$\therefore \frac{dv}{d\lambda} = f(k,\ N_e,\ K_7) \qquad (1.54)$$

From equation (1.38),

$$\frac{dF(\lambda)}{d\lambda} = \frac{v du/d\lambda - u dv/d\lambda}{v^2 + u^2}$$

and from equations (1.51) to (1.54) inclusive, it can be seen that

$$\frac{dF(\lambda)}{d\lambda} = f(\phi_L,\ k,\ N_e,\ K_7)\ .\ f(\text{cutter}) \qquad (1.55)$$

In order to examine the convergence of equation (1.38) as defined by equation (1.55), when applied to an ISO 53 (6) cutter, a computer programme was written, independent of gear design, to encompass four nested DO-LOOPS of the variables K_7, k, N_e and ϕ_L.

From Figure 1.8, it can be seen that K_7 can only vary from 0 to 0.5π radians. However, in order to examine the essence of the function, K_7 was varied from -0.5π to $+0.5\pi$, in increments of 0.01, constituting 317 loops.

Backlash was set at zero, and hence the addendum modification coefficient, x, was utilised in lieu of k, which was varied from - 0.5 to + 1.0, in increments of 0.1. These limits were set to correspond to the conventional range of ISO/TR 4467(7), and constituted 16 loops.

The virtual number of teeth, N_e, was set as a sixty element array as follows:

10, 11, 12, 13, 14, 15, 16, 17, 18, 19, 20, 22, 24, 26,
28, 30, 32, 34, 36, 38, 40, 42, 44, 46, 48, 50, 52, 54,
56, 58, 60, 65, 70, 75, 80, 85, 90, 95, 100, 105, 110,
120, 130, 140, 150, 160, 170, 180, 190, 200, 300, 400,
500, 1000, 1400, 2000, 3000, 5000, 10000, 10000000

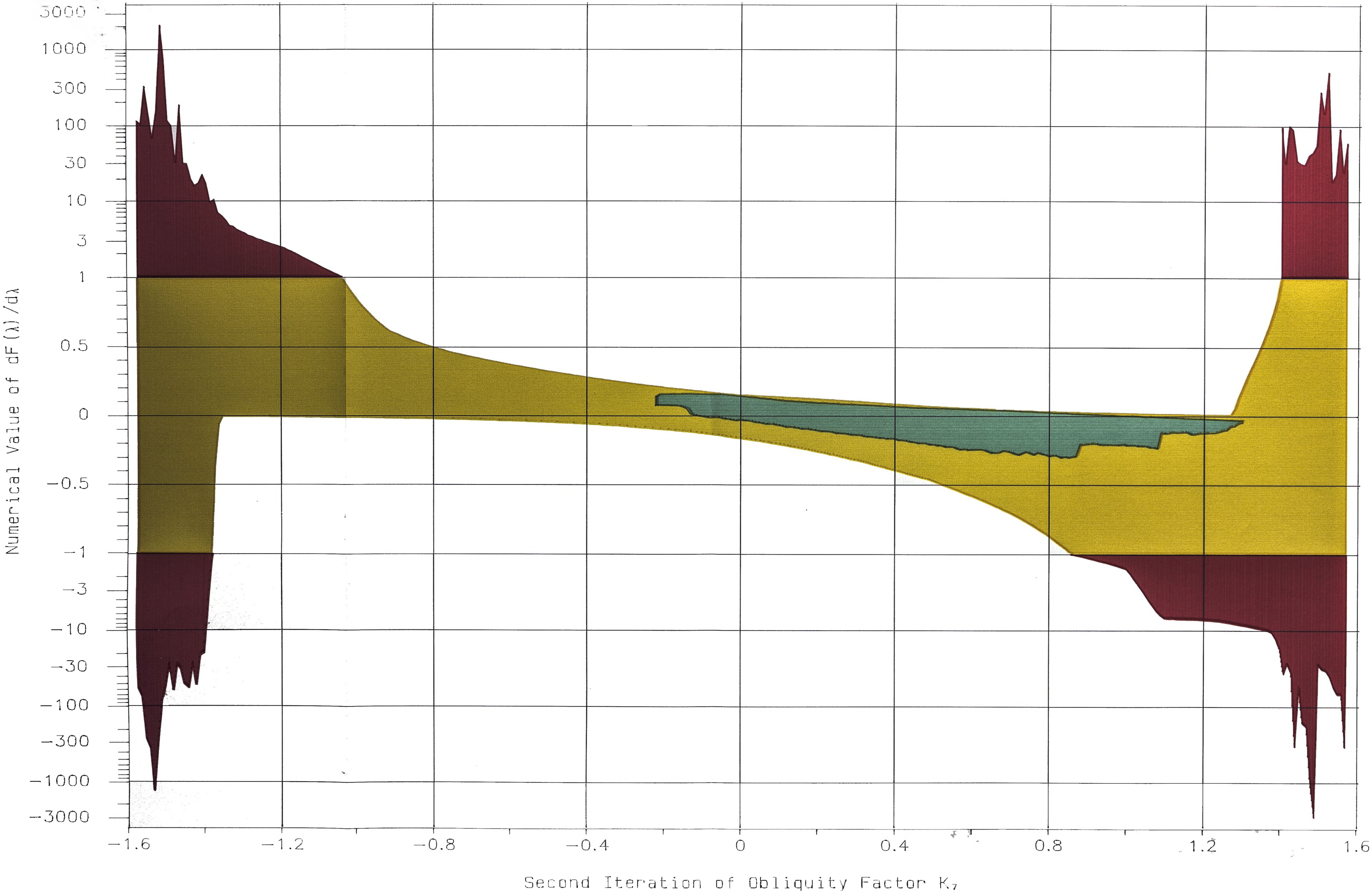


FIGURE 1.9 – CONVERGENCE OF THE LAMBDA METHOD WITHIN THE LIMITS OF ISO/TR4467

The load angle, ϕ_L, was varied from 0.0 to 1.0 radians in increments of 0.1, representing 11 loops.

Hence the total number of combinations generated was 317x16x60x11 or 3,347,520 of which 180,669 were non-convergent. The maximum and minimum values of $dF(\lambda)/d\lambda$ were recorded for a particular value of K_7. These values are tabulated in APPENDIX B, and are plotted in red on Figure 1.9, whilst the yellow bandwidth indicates the finite range of the 3,166,851 examples where the $\left| dF(\lambda)/d\lambda \right|$ was less than one; ie, for which the function converged. Further, the highly transcendental nature of equation (1.38) is clearly demonstrated.

The problem now remained to examine combinations of the variables K_7, k, N_e and ϕ_L when applied to "real gears", where a "real gear" was defined as any gear that could be cut, no matter how incongruous the final tooth shape may have been. In keeping with this criterion, the following tests were applied to the relevant sections of the program:

i. The conventional addendum modification coefficient limits of Clause 3.4.2 of ISO/TR 4467-1982(E) were applied. This is in agreement with the tenet of AS2938-1987. However, it is appreciated that gears with addendum modification coefficients outside of this range, may be subjected to analysis by the application of the Lambda Method. This possibility is the subject of further discussion in Section 9.2 - Suggestions for Further Study.

ii. Utilising an ISO 53 cutter, and for a particular combination of K_7, k and N_e, then from equation (1.41) there is a unique value of lambda λ_o,

$$\lambda_o = 0.5\pi - 2K_2/N_e - K_7 - \tan(K_7)/K_3 \qquad (1.41a)$$

where $K_2 = f$ (cutter)

$K_3 = f\ (N_e, K_1)$

$K_1 = f$ (cutter, k)

From Figure 1.8 it can be seen that λ_o must be within the range $0 < \lambda_o < 0.5\pi$.

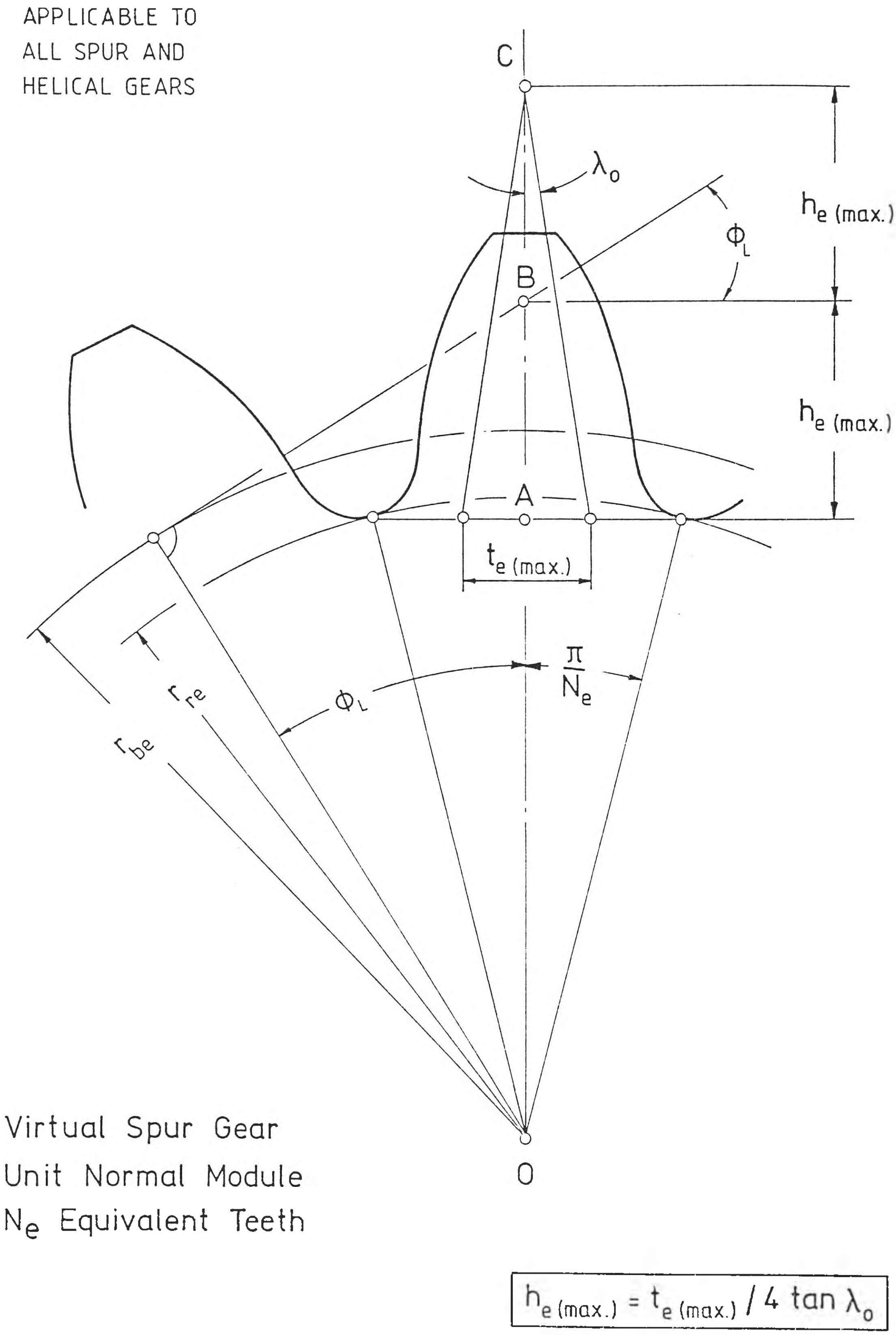


FIGURE 1.10 - MAXIMUM PARABOLA DIMENSIONS

iii. From Figure 1.10, it can be seen that for a particular combination of K_7, k, N_e and ϕ_L, there is a unique value of λ_o as defined by equation (1.41a) and a unique value of $h_{e(max)}$.

Now $$OB = r_{be} \sec(\phi_L) \tag{1.56}$$

and $$OA = r_{re} \cos(\pi/N_e) \tag{1.57}$$

$$\therefore h_{e(max)} = r_{be} \sec(\phi_L) - r_{re} \cos(\pi/N_e) \tag{1.58}$$

For the particular combination of K_7, k, N_e and ϕ_L, the value of h_{eo} is calculated from equation (1.25).

If $h_{eo} > h_{e(max)}$ or $h_{eo} \leq 0.0$, then h_{eo} could be rejected.

Similarly, $$t_{e(max)} = 4\ h_{e(max)} \tan(\lambda_o) \tag{1.59}$$

The value of t_{eo} was calculated from equation (1.26).

If $t_{eo} > t_{e(max)}$ or $t_{eo} \leq 0.0$ then t_{eo} could be rejected.

When the three tests were applied to the original programme, the number of combinations was reduced from 3,347,520 to 804,776. The first test rejected 75,446 combinations which were outside the conventional limits of ISO/TR 4467, whilst the second test rejected 122,177 combinations where $0.5\pi \leq \lambda_o \leq 0.0$.

A further 399,073 combinations were rejected where $t_{eo} > t_{e(max)}$.

Lastly, 130 combinations were rejected where $h_{eo} \leq 0.0$. Hence, 207,950 combinations were accepted, all of which converged.

The maximum and minimum values of $dF(\lambda)/d\lambda$ were recorded for a particular value of K_7. These values are tabulated in APPENDIX B, and are plotted in green on Figure 1.9, indicating that any gear manufactured with an ISO 53 cutter, and having addendum modifications within the conventional limits of ISO/TR 4467, will converge, when analysed by the Lambda Method.

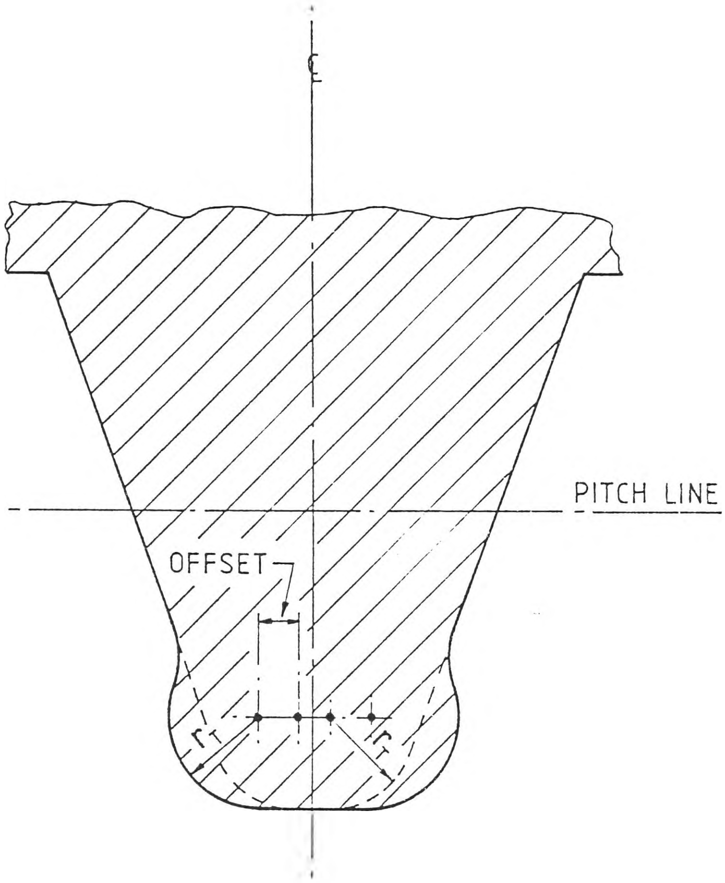


FIGURE 1.11 - PROTUBERANCE HOB

1.7 Interpretation of the Lambda Method

Prior to the development of any analytical method, one must define the cutter by a series of equations. This then poses the problem of cutters with tip relief and protuberance.

Referring to Figure 1.6, tip relief causes the load line to move towards the centre-line of the tooth and hence raises the point f, the apex of the parabola. However, a rigorous mathematical investigation of this phenomenon within the limits of recommended tip relief for standard cutters, led the author to the conclusion that the effect of tip relief on the parabola dimensions could be neglected.

Protuberance has a dramatic effect on the shape of the trochoid and consequently cannot be neglected. However, the problem is to mathematically define protuberance. One approach which has been used in previous analytical methods (8) is shown in Figure 1.11.

Here it has been assumed that the protuberance is tangential to the outside diameter of the cutter, circular and offset by some specified amount. The blending of the circular protuberance into the flank of the cutter is assumed not to influence the shape of the trochoid. Whilst this approach can be defined mathematically, it assumes that the final tooth shape will not change after grinding.

It is the author's experience that the shape of the protuberance varies greatly, and is usually an "in house" design based on the manufacturer's gear grinding facilities. In fact, the tendency internationally is to head in the direction of protuberance hobs, which after gear grinding, produce a tooth profile as if it had been hobbed with a standard cutter, thus removing the undercut usually associated with the use of protuberance hobs.

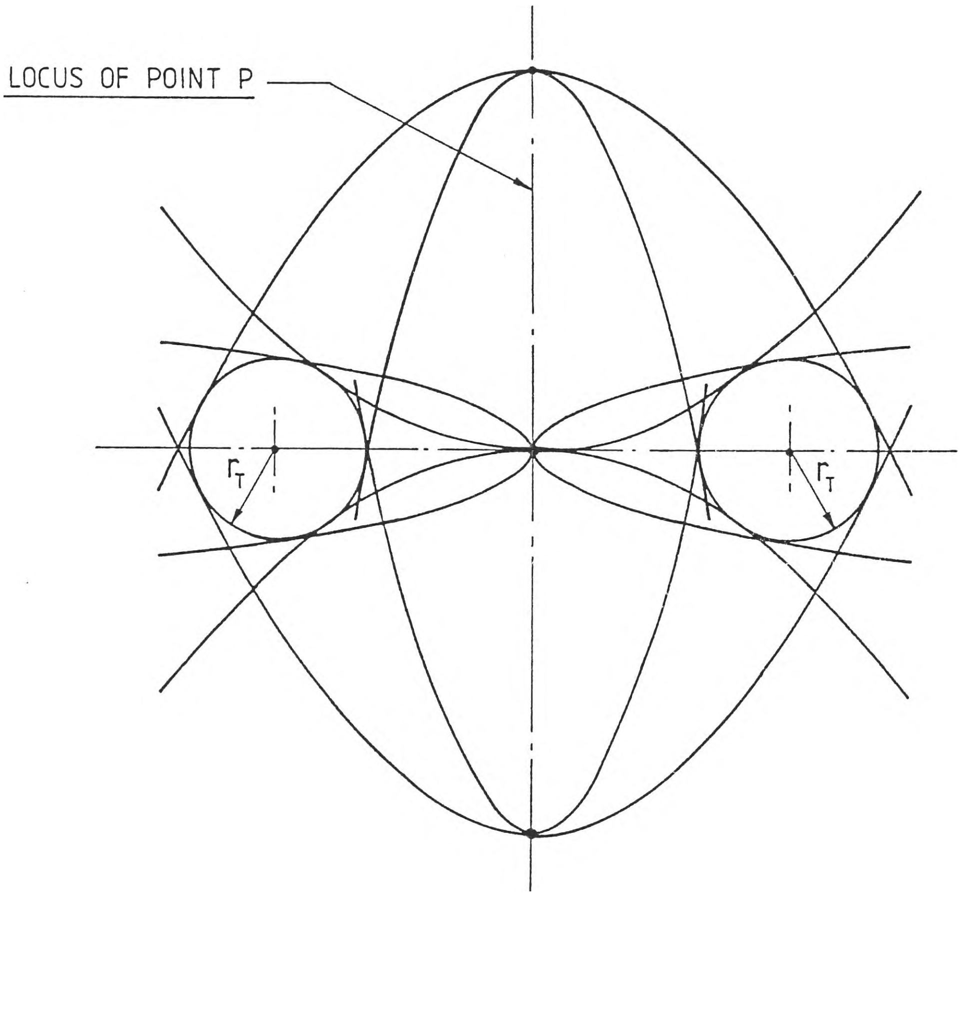


FIGURE 1.12 - MATHEMATICAL PARABOLAS

Due to the "carte blanche" approach to the design of protuberance hobs, the author believes that the inclusion of protuberance hobs in a generalised analytical method for the determination of the parabola dimensions could lead to potential problems.

For gears cut with protuberance hobs, the author suggests that a CAD tooth profile be drawn, utilising the actual shape of the protuberance hob employed, and then modified to account for the grinding.

However, should it be concluded that the protuberance profile as identified in Figure 1.11 is acceptable, then this definition of protuberance can be incorporated into the analytical method as follows.

Referring to Section 1.3, Rack Cutter Constants, equation (1.9) defines the horizontal distance, K_2, of the centre of the rack tip radius from the centre-line of the rack. If this distance is reduced by the "offset" of the protuberance as defined in Figure 1.11 then the analytical method presented in Section 1.5 is still applicable to protuberance hobs.

The highly transcendental nature of the analytical method as shown in Figure 1.9, can be elucidated if the mathematics involved in the determination of the Lewis parabola are isolated from gear design and redefined as follows: "Two circles of radius r_T are drawn such that their centres are equidistant from a point P. Draw a parabola that is tangential to the circles and has its apex at point P".

As the equations of the circle and parabola are both functions of squared terms, a fourth order equation is generated, yielding four possible solutions. This phenomenon is shown diagrammatically in Figure 1.12, which lends itself to the logo shown on the flyleaf.

Section 1.6 has shown that the absolute value of the dependent variable lambda, $|\lambda|$, when returned to the iteration procedure will always converge and, at the same time, select the correct parabola.

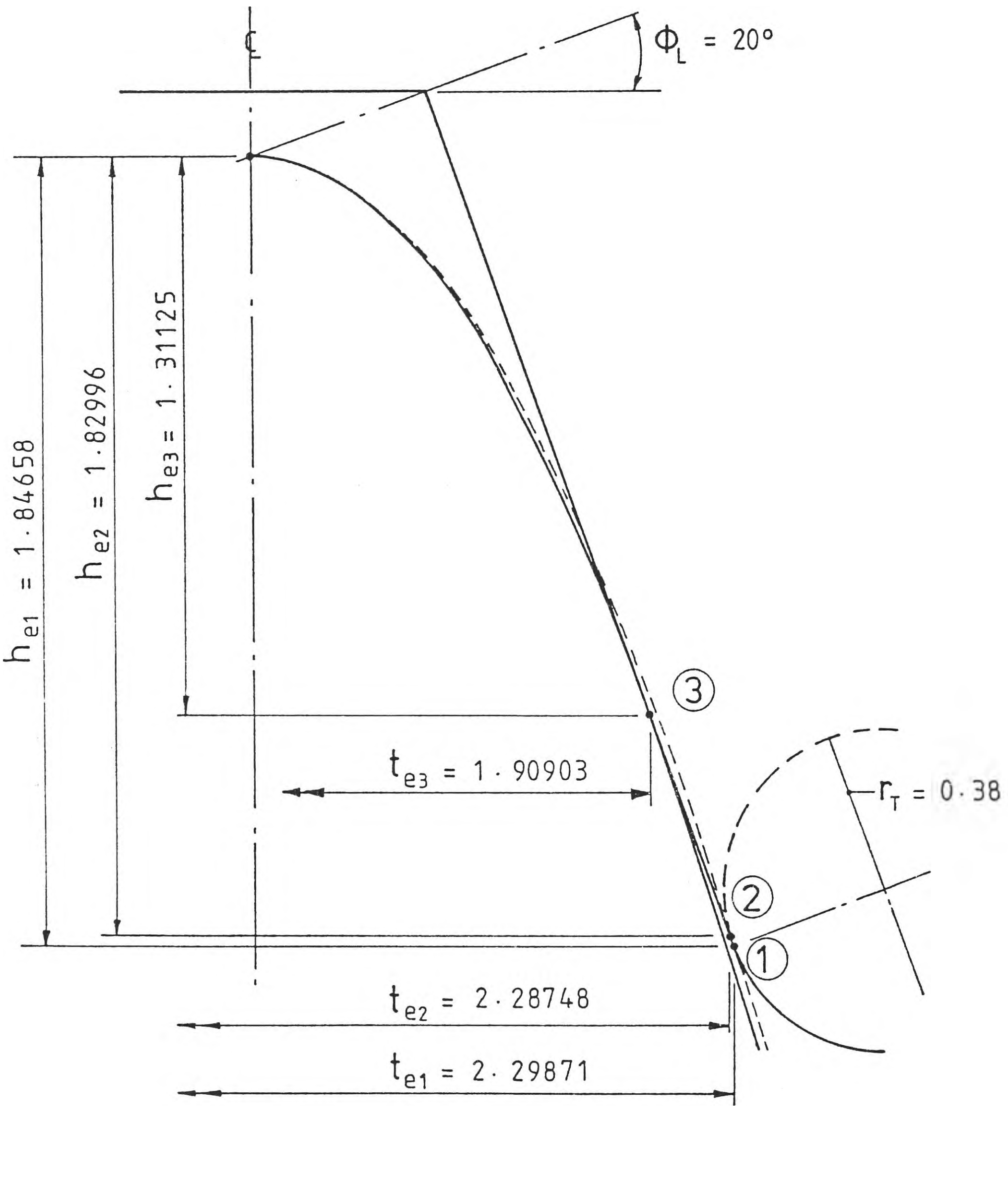


FIGURE 1.13 - LEWIS PARABOLAS

However, having proved the convergence of the mathematical iteration, it still remains to define convergence when applied to the Lewis parabola as utilised in gear design. For gears with large tooth numbers, and/or severely modified (approaching all addendum) teeth, it may not be possible to inscribe a parabola that is tangent to the root fillet. This phenomenon is best explained by way of an example. Consider the calculation of the strength geometry factor of a rack with tip loading. The analysis of a rack has the advantage that all parabola dimensions can be calculated from analytical geometry.

An ISO 53 unit module spur rack is shown in Figure 1.13, with three possible interpretations of the convergence of the Lewis parabola. Firstly, if it is assumed that point 1 (the intersection point of the trochoidal fillet curve and the tooth involute), is the correct interpretation, then $t_{e1} = 2.29871$ and $h_{e1} = 1.84658$. Secondly, if point 2 (the point of tangency of the parabola and the continuation of the trochoidal fillet), is the correct interpretation, then $t_{e2} = 2.28748$ and $h_{e2} = 1.82996$. However, it will be observed that the parabolas passing through points 1 and 2, pass outside the tooth profile, and hence a third interpretation that the parabola must be inscribed within the tooth profile is presented, yielding $t_{e3} = 1.90903$ and $h_{e3} = 1.31125$. It is of interest to note that the subsequent calculations of the AGMA 218.01 strength geometry factors are respectively $J_1 = 0.3173$, $J_2 = 0.3169$ and $J_3 = 0.3009$.

The author suggests that due to the nature of the calculation of the stress concentration factor at the root fillet in AGMA 218.01, then point 2 is the correct interpretation of the convergence of the Lewis parabola. However, should it be decided that point 1 is the correct interpretation, then the radius to the top of the trochoid, R_F, can be accurately calculated from Buckingham (9). The chordal height and thickness can be calculated for any radius on the involute and hence the parabola width equals the chordal thickness, the chordal height being modified to account for the difference between the outside radius of the gear and the apex of the parabola. If it is concluded that point 3 is the point of convergence of the Lewis parabola, then a series of equations such as those presented will be inapplicable.

However, a computer program for protuberance hobs such as that developed by Davey and Wilkinson (10) will locate point 3.

The equations presented in this Thesis yield the parabola height and width as defined by point 2. They have been tested with $1x10^{10}$ wheel teeth to yield the parabola dimensions of Figure 1.13, which were calculated from analytical geometry.

Prior to the acceptance of any analytical method for the determination of strength geometry factors, the problems of a mathematical definition of protuberance and the correct interpretation of the convergence of the Lewis parabola must be resolved.

The author believes that protuberance cannot be satisfactorily defined to encompass all forms of protuberance and the subsequent modification to the tooth profile as a result of grinding or shaving. The shape of the final tooth profile is significant, as can be demonstrated by the effect that the inclusion of backlash has in the calculation of the strength geometry factor. In fact, the inclusion of backlash has a greater effect on the numerical value of the strength geometry factor than does several steps of iteration.

The validity of the debate as to whether point 1 or point 2 of Figure 1.13, is the correct interpretation of the convergence of the Lewis parabola is somewhat academic as the effect on the strength geometry factor is relatively insignificant. However, any analytical method should yield the correct values for the parabola dimensions. Once the point of convergence is resolved, the analytical method presented can accommodate either interpretation.

The simplicity, coupled with the degree of accuracy and the ability to accommodate LCR helical gears, has led the Standards Association of Australia to adopt the analytical method described in this Thesis as the approved method for the calculation of geometry factors in the Australian Standard for Gear Rating AS 2938, which is based on AGMA 218.01.

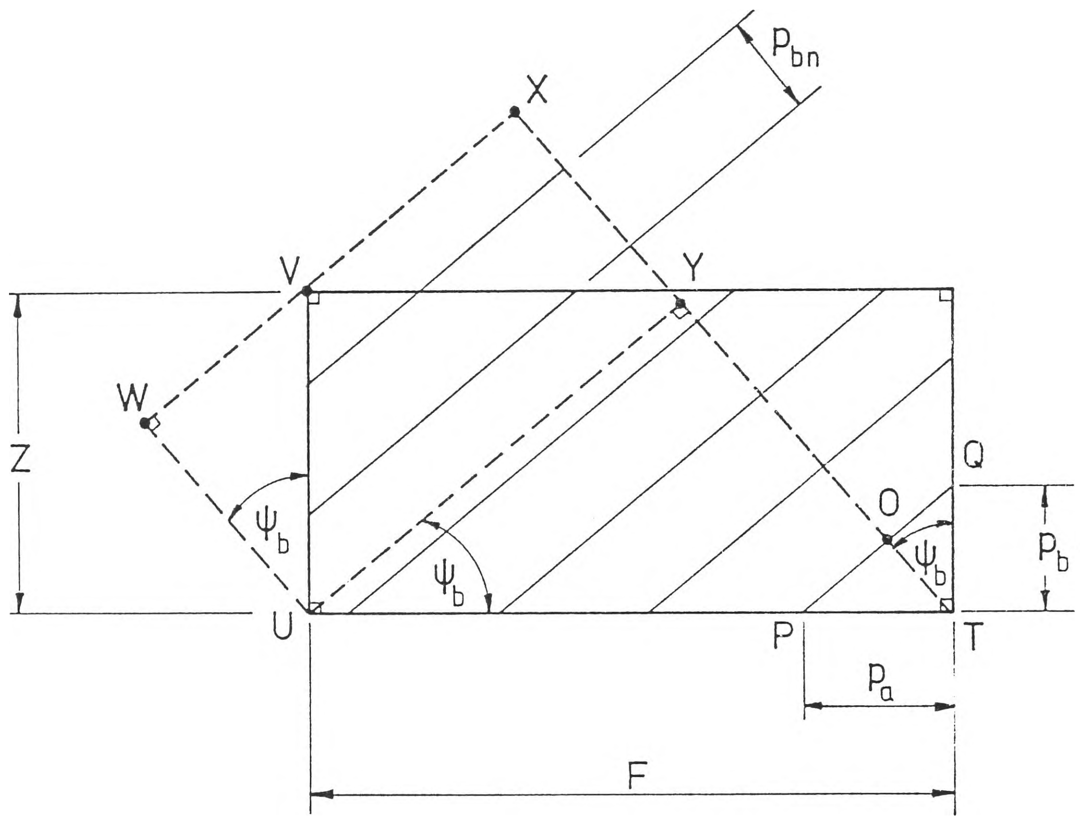


FIGURE 1.14 - LIMITING NUMBER OF LINES OF CONTACT

1.8 Length of the Lines of Contact

Prior to the procedual steps for the calculation of the geometry factors of AGMA 218.01, an equation is derived, for the minimum length of the lines of contact for helical gears.

The analysis of the length of the lines of contact for helical gears is achieved by an examination of the contact zone, this being the projection of the contact path over the facewidth of the gear.

The following equation is presented as an alternative to Section 6.2.6 of AGMA 218.01.

Referring to Figure 1.14, assume that the teeth are so placed that one line of contact, of zero length, passes through point T. The limiting number of lines of contact, n, then equals the length of the line TX, normal to the lines of contact, divided by the normal base pitch, truncated to the next lowest integer value.

From Δ TYU

$$TY = F \sin(\psi_b)$$

From Δ UVW

$$UW = Z \cos(\psi_b)$$

and $$XY = UW$$

$$\therefore \quad TX = F \sin(\psi_b) + Z \cos(\psi_b) \tag{1.60}$$

From Δ TOQ

$$TO = TQ \cos(\psi_b)$$

$$\therefore \quad p_{bn} = p_b \cos(\psi_b) \tag{1.61}$$

From equations (1.60) and (1.61),

$$n = \mathrm{INT}\left[\frac{F \sin(\Psi_b) + Z \cos(\Psi_b)}{p_b \cos(\Psi_b)}\right]$$

ie, $$n = \mathrm{INT}\left[\frac{F \tan(\Psi_b)}{p_b} + \frac{Z}{p_b}\right] \tag{1.62}$$

From Δ TPQ

$$TP = TQ/\tan(\Psi_b)$$

$$\therefore \quad p_a = p_b/\tan(\Psi_b) \tag{1.63}$$

From equations (1.62) and (1.63),

$$n = \mathrm{INT}\left[\frac{F}{p_a} + \frac{Z}{p_b}\right] \tag{1.64}$$

By definition, the face contact ratio, m_F, is equal to the facewidth divided by the axial pitch.

$$\therefore \quad m_F = F/p_a \tag{1.65}$$

Similarly, the transverse contact ratio, m_p, is equal to the length of the line of contact, divided by the base pitch.

$$\therefore \quad m_p = Z/p_b \tag{1.66}$$

From equations (1.64), (1.65) and (1.66),

$$n = \mathrm{INT}(m_F + m_p) \tag{1.67}$$

Having established the limiting number of the lines of contact, the problem remains to calculate the total length of these lines, L_{min}, contained within the rectangle bounded by the facewidth, F, and the length of the line of contact, Z.

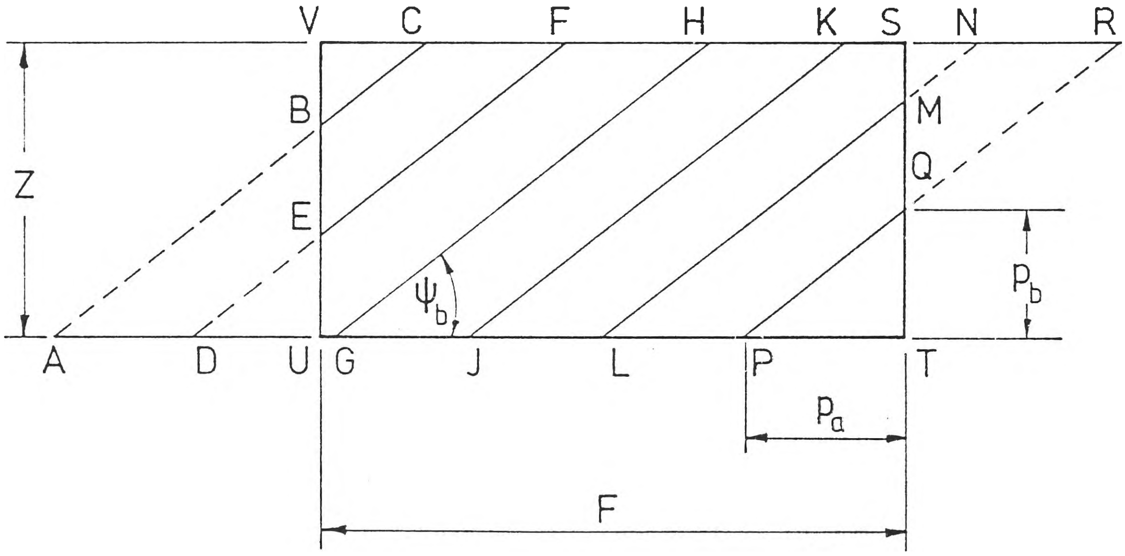


FIGURE 1.15 - MINIMUM LENGTHS OF LINES OF CONTACT

Referring to Figure 1.15, the total length of the lines AC, DF, GH, JK, LN and PR is

$$L = n\ Z\ /\ \sin(\psi_b) \tag{1.68}$$

In order to calculate the total length of the lines contained with the rectangle, the lengths of the lines AB, DE, MN and QR must be subtracted from L.

Now $\quad SQ = Z - p_b$

$$\therefore \quad QR = (Z - p_b)/\sin(\psi_b) \tag{1.69}$$

Similarly, $SM = Z - 2\ p_b$

$$\therefore \quad MN = (Z - 2\ p_b)/\sin(\psi_b) \tag{1.70}$$

This procedure would continue until no contact lines project beyond the boundary ST, ie, until the value of Z/p_b is truncated to the next lowest integer. Thus,

$$\text{Limit} = [Z - \text{INT}(Z/p_b)\ p_b]/\sin(\psi_b) \tag{1.71}$$

Consider the calculation of the length of the line DE.

The length of the line AT is equal to an integer multiple of the axial pitch p_a, where the integer is equal to the limiting number of lines of contact n.

$$\therefore \quad AT = n\ p_a \tag{1.72}$$

From equation (1.65), the facewidth, F, can also be expressed as a function of p_a.

$$F = m_F\ p_a \tag{1.73}$$

From equations (1.72) and (1.73),

$$AU = (n - m_F)\ p_a \tag{1.74}$$

Now the numerical value of $n - m_F$ will contain an integer portion and a fractional portion. As AD is an integer of the axial pitch, then the length DU must correspond to the fractional portion of $n - m_F$.

ie, $$DU = FRAC(n - m_F)\ p_a \tag{1.75}$$

$$\therefore \quad DE = FRAC(n - m_F)\ p_a/\cos(\Psi_b) \tag{1.76}$$

Similarly, AU is equal to the fractional portion of $n - m_F$, plus one axial pitch.

$$AU = [FRAC(n - m_F) + 1\ ]\ p_a \tag{1.77}$$

$$\therefore \quad AB = [FRAC(n - m_F) + 1\ ]\ p_a/\cos(\Psi_b) \tag{1.78}$$

This procedure would continue until no contact lines project beyond the boundary UV, ie, until the integer value of $n - m_F$ is reached.

Thus $$\text{Limit} = [FRAC(n - m_F) + INT(n - m_F)]\ p_a/\cos(\Psi_b) \tag{1.79}$$

It should be noted that if $m_F > n$, then no lines of contact project beyond the boundary UV, and hence $n - m_F$ is set equal to zero.

Now combining equations (1.68), (1.69), (1.70), (1.71), (1.76), (1.78) and (1.79) to obtain a general equation for the minimum length of the lines of contact, and letting $C_o = n - m_F$ in combination with rearranging equation (1.63) to:

$p_a/\cos(\Psi_b) = p_b/\sin(\Psi_b)$ gives

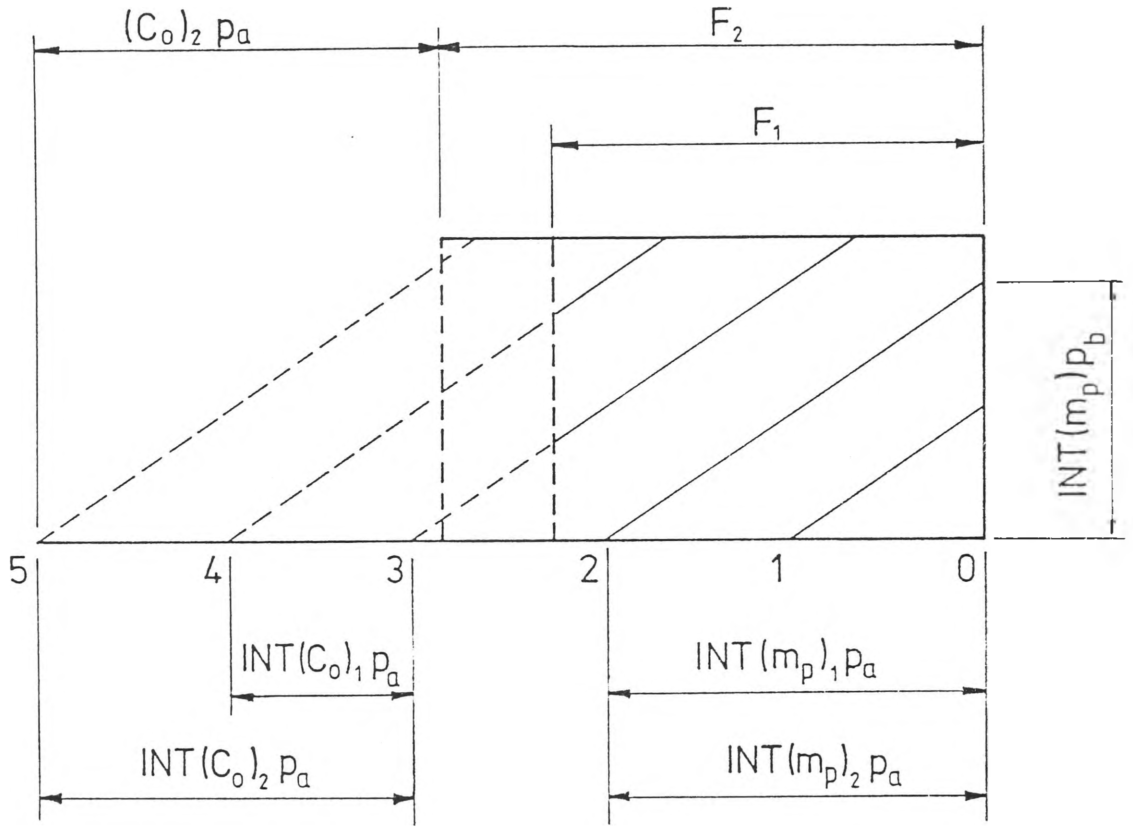


FIGURE 1.16 - VARIATION OF C_0 AS A FUNCTION OF FACE WIDTH

$$L_{min} = \frac{n\,Z}{\sin(\Psi_b)} - \frac{Z - p_b}{\sin(\Psi_b)} - \frac{Z - 2p_b}{\sin(\Psi_b)} \;\ldots\ldots\ldots - \frac{Z - \mathrm{INT}(m_p)p_b}{\sin(\Psi_b)}$$

$$- \frac{\mathrm{FRAC}(C_o)p_b}{\sin(\Psi_b)} - \frac{[\,\mathrm{FRAC}(C_o) + 1\,]p_b}{\sin(\Psi_b)}$$

$$\ldots\ldots\ldots - \frac{[\,\mathrm{FRAC}(C_o) + \mathrm{INT}(C_o)\,]p_b}{\sin(\Psi_b)}$$

$$L_{min} = \mathrm{cosec}(\Psi_b)\ \{\ n\,Z - Z + p_b - Z + 2p_b \;\ldots\ldots - Z + \mathrm{INT}(m_p)p_b$$

$$- p_b\ [\ \mathrm{FRAC}(C_o) + \mathrm{FRAC}(C_o) + 1 \ldots + \mathrm{FRAC}(C_o) + \mathrm{INT}(C_o)\ ]\ \}$$

ie,

$$L_{min} = \mathrm{cosec}(\Psi_b)\ \{\ n\,Z - \mathrm{INT}(m_p)\,Z + p_b\ [\ \sum_{k=1}^{\mathrm{INT}(m_p)} k\ ]$$

$$- p_b\ [\ \mathrm{INT}(C_o + 1).\mathrm{FRAC}(C_o) + \sum_{k=1}^{\mathrm{INT}(C_o)} k\ ]\ \}$$

From equation (1.66)

$$Z = m_p\ p_b$$

$\therefore$

$$L_{min} = p_b\ \mathrm{cosec}(\Psi_b)\ \{\ m_p\ [n - \mathrm{INT}(m_p)] - \mathrm{INT}(C_o + 1).\mathrm{FRAC}(C_o)$$

$$+ \sum_{k=1}^{\mathrm{INT}(m_p)} k - \sum_{k=1}^{\mathrm{INT}(C_o)} k\ \} \qquad (1.80)$$

Consider $\mathrm{INT}(m_p) - \mathrm{INT}(C_o)$

Referring to Figure 1.16, it can be seen that when the facewidth is equal to F_1,

$$\mathrm{INT}(m_p)_1 - \mathrm{INT}(C_o)_1 = 1$$

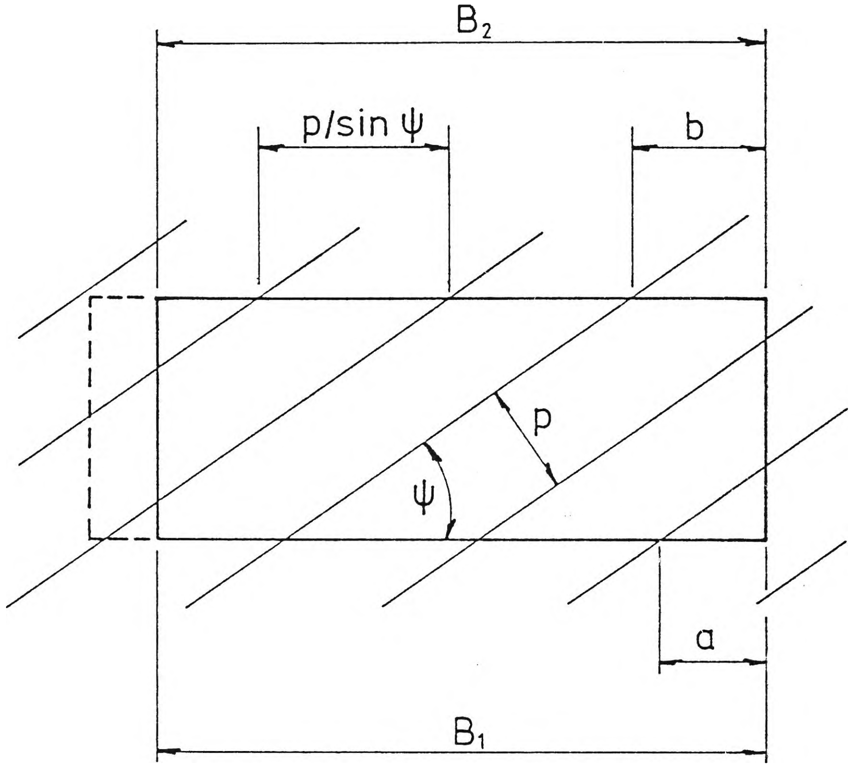


FIGURE 1.17 - VARIATION IN THE NUMBER OF INTERSECTING LINES

Similarly, when the facewidth is equal to F_2,

$$INT(m_p)_2 - INT(C_o)_2 = 0$$

This observation can be expressed in the axiom, "the number of equally spaced parallel lines drawn through a rectangle, are so distributed that the number of lines intersecting two parallel sides of the rectangle, differs by a maximum of one".

To prove the axiom, consider Figure 1.17. The maximum number of lines n, that intersect the side B_1 is

$$n_1 = INT[(B_1 - a)\sin(\psi)/p)] + 1 \qquad (1.81)$$

Similarly,

$$n_2 = INT[(B_2 - b)\sin(\psi/p)] + 1 \qquad (1.82)$$

Combining equations (1.81) and (1.82) and noting that B_1 equals B_2 for a rectangle gives

$$n_1 - n_2 = INT\left[\frac{B_1 \sin(\psi)}{p} - \frac{a\sin(\psi)}{p}\right] - INT\left[\frac{B_1\sin(\psi)}{p} - \frac{b\sin(\psi)}{p}\right]$$

Now, $0 \leq a \leq \dfrac{p}{\sin(\psi)}$ and $0 \leq b \leq \dfrac{p}{\sin(\psi)}$

$$\therefore \quad 0 \leq \frac{a\sin(\psi)}{p} \leq 1 \quad \text{and} \quad 0 \leq \frac{b\sin(\psi)}{p} \leq 1$$

Hence, the maximum absolute difference between

$\dfrac{a\sin(\psi)}{p}$ and $\dfrac{b\sin(\psi)}{p}$ is 1

Rearranging equation (1.80) and incorporating the axiom, gives

$$\sum_{k=1}^{INT(m_p)} k - \sum_{k=1}^{INT(C_o)} k = 1 + 2 + \ldots + INT(C_o) + INT(m_p)$$

$$- 1 - 2 - \ldots. - INT(C_o)$$

$$= INT(m_p) \text{ if } INT(m_p) > INT(C_o)$$

$$= 0 \text{ if } INT(m_p) = INT(C_o)$$

$$\sum_{k=1}^{INT(m_p)} k - \sum_{k=1}^{INT(C_o)} k = INT(m_p)[INT(m_p)-INT(C_o)] \quad (1.83)$$

From equations (1.80) and (1.83)

$$L_{min} = p_b \{ m_p [n - INT(m_p)] - INT(C_o + 1) FRAC(C_o)$$

$$+ INT(m_p) [INT(m_p) - INT(C_o)]\}/ \sin(\psi_b) \quad (1.84)$$

Now $INT(C_o + 1) = INT(C_o) + 1$

$$\therefore \quad INT(C_o+1).FRAC(C_o) = INT(C_o).FRAC(C_o) + FRAC(C_o)$$

$$= INT(C_o).FRAC(C_o) + C_o-INT(C_o) \quad (1.85)$$

Similarly,

$$INT(m_p) [INT(m_p) - m_p] = - INT(m_p).FRAC(m_p) \quad (1.86)$$

From equations (1.84), (1.85) and (1.86),

$$L_{min} = p_b [m_p n - C_o - INT(m_p).FRAC(m_p)$$

$$- INT(C_o)\{INT(m_p) + FRAC(C_o) - 1\}]/\sin(\psi_b) \quad (1.87)$$

The complete sequence for the calculation of the minimum length of the lines of contact is given in Section 1.9.

1.9 Procedural Steps for the Calculation of the Geometry Factors of AGMA 218.01 using the Lambda Method

This Section sets out mathematical procedures for calculating the value of the geometry factors for both surface durability I, and beam strength J. Any straight sided basic rack without protuberance and any combination of addendum modification coefficients, topping and backlash allowance can be treated. The procedure is capable of adaption for use on computers (refer to Appendix A) or programmable calculators.

Note: 1. Where possible, symbols agreeing with AGMA 112.04 (11) and AGMA 218.01 have been selected. Some quantities which have to be calculated have no specific symbols in either AGMA Code, and for these, symbols of the general type favoured by AGMA 112.04 have been utilised and defined throughout the text.

2. It is assumed that the following quantities are known:

N_P = Number of pinion teeth.

N_G = Number of wheel teeth.

ϕ_c = Normal profile angle of the equivalent standard rack cutter.

m_n = Standard normal metric module.

Ψ_s = Helix angle at standard pitch diameter.

x_P = Addendum modification coefficient of pinion with zero backlash.

x_G = Addendum modification coefficient of wheel with zero backlash.

F = Net facewidth of the narrowest member.

B_{NP} = Backlash applied to the pinion in the normal plane.

B_{NG} = Backlash applied to the wheel in the normal plane.

Δr_o = Radial tooth topping applied to the pinion.

ΔR_o = Radial tooth topping applied to the wheel.

r_T = Edge radius of the cutting tool.

h_a = Standard addendum of the basic rack.

h_b = Standard dedendum of the basic rack.

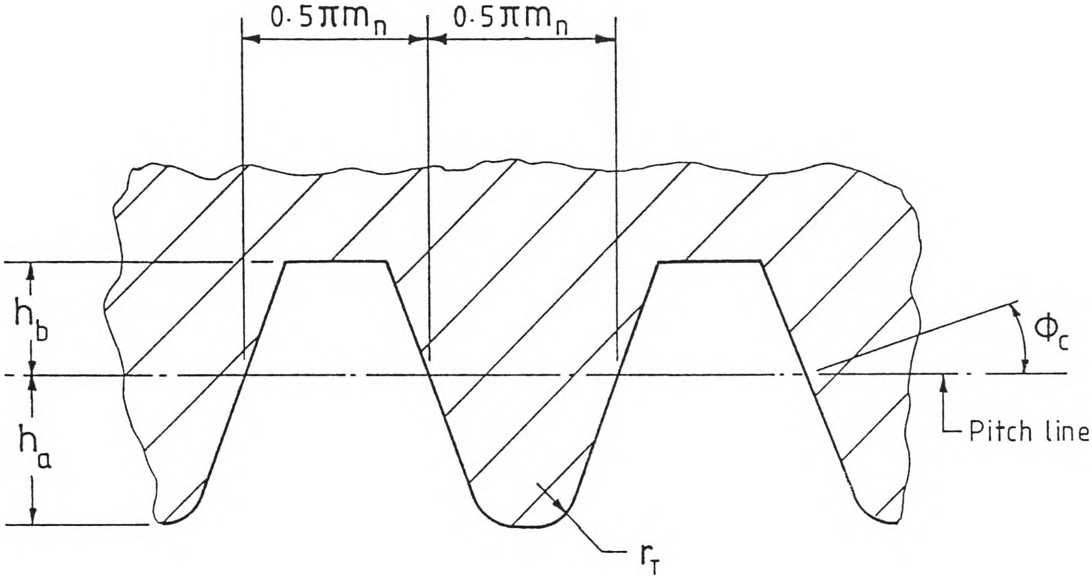


FIGURE 1·4A - BASIC CUTTING TOOL DEFINITIONS

3. Normal tip relief has a negligible effect on the final calculated value of "J".

4. Normal backlash has a significant effect on the final calculated value of "J".

5. L_{min} has a significant effect on the final calculated value of "J".

6. The calculation is applicable to external spur, LCR and conventional helical gears provided only that N_P is never considered to be greater than N_G. For internal gearing refer to AGMA 218.01.

7. All angular calculations are in radians.

8. For examples of calculation refer to APPENDIX C.

The following 32 steps are required to calculate I and J.

1. Calculate the transverse pressure angle at standard diameter ϕ_s

$$\phi_s = \tan^{-1}(\tan(\phi_c)/\cos(\psi_s))$$

Note: For spur gears $\phi_s = \phi_c$

2. Calculate the operating transverse pressure angle ϕ_t

$$\mathrm{inv}\,(\phi_t) = 2\tan(\phi_c)(x_P + x_G)/(N_P + N_G) + \mathrm{inv}(\phi_s)$$
$$\phi' = \phi_s - [\tan(\phi_s) - \phi_s - \mathrm{inv}(\phi_t)]/\tan^2(\phi_s)$$
$$\phi'' = \phi' - [\tan(\phi') - \phi' - \mathrm{inv}(\phi_t)]/\tan^2(\phi')$$
$$\phi_t = \phi'' - [\tan(\phi'') - \phi'' - \mathrm{inv}(\phi_t)]/\tan^2(\phi'')$$

Note:

a. If $x_P + x_G = 0$ then $\phi_t = \phi_s$

b. $\mathrm{inv}(\phi) = \tan(\phi) - \phi$

3. Calculate the operating centre distance C

$$C = 0.5\, m_n\,(N_P + N_G)\sec(\psi_s)\cos(\phi_s)\sec(\phi_t)$$

4. Calculate the base radii r_b and R_b

$$r_b = 0.5\, N_P\, m_n \sec(\psi_s)\cos(\phi_s)$$

$$R_b = r_b\, N_G / N_P$$

5. Calculate the operating pitch radii r and R

$$r = N_P\, C/(N_P + N_G)$$

$$R = C - r$$

6. Calculate the tip radii r_o and R_o

$$r_o = m_n(0.5\, N_P \sec(\psi_s) + x_P) + h_b - \Delta r_o$$

$$R_o = m_n(0.5\, N_G \sec(\psi_s) + x_G) + h_b - \Delta R_o$$

7. Calculate the face contact ratio m_F

$$m_F = F \sin(\psi_s)/(\pi \, m_n)$$

Note:

a. For spur gears $m_F = 0$

b. If $m_F \leq 1$ and $\psi_s \neq 0$ then the gears are classified as low contact ratio (LCR) helical gears

c. If $m_F > 1$ and $\psi_s \neq 0$ then the gears are classified as conventional helical gears

d. For herringbone and double helical gears F is the width of one helix only

e. It is suggested that where possible, m_F should be greater than 1.0 if $\psi_s \neq 0$. ie. $F > \pi \, m_n \,/\, \sin(\psi_s)$

8. Calculate the length of the approach and recess paths Z_b and Z_a respectively,

$$Z_b = [R_o^2 - R_b^2]^{0.5} - [R^2 - R_b^2]^{0.5}$$

$$Z_a = [r_o^2 - r_b^2]^{0.5} - [r^2 - r_b^2]^{0.5}$$

9. Calculate the length of the line of action in the transverse plane Z

$$Z = Z_b + Z_a$$

10. Calculate the distance along the line of action from the pitch point to the point of stress calculation Z_c

a. For spurs and LCR helicals with $m_F \leq 1.0$

$$Z_c = \pi \, m_n \sec(\psi_s) \cos(\phi_s) - Z_a$$

b. For conventional helical gears with $m_F > 1.0$

$$Z_c = [r^2 - r_b^2]^{0.5} - [0.25 \, (C - R_o + r_o)^2 - r_b^2]^{0.5}$$

11. Calculate the contact height factor C_x

$$C_x = (\sin(\phi_t) - Z_c / r)(\sin(\phi_t) + Z_c / R)/\sin^2(\phi_t)$$

Note:

For spurs and LCR helicals Z_c is calculated from step 10a.
For conventional helical gears Z_c is calculated from step 10b.

12. <u>For LCR helical gears only</u> with $0 < m_F \leq 1.0$ calculate the contact height factor C_{xh}

$$C_{xh} = (\sin(\phi_t) - Z_{ch} / r)(\sin(\phi_t) + Z_{ch} / R) / \sin^2(\phi_t)$$

where

$$Z_{ch} = [r^2 - r_b^2]^{0.5} - [0.25\ (C - R_o + r_o)^2 - r_b^2]^{0.5}$$

13. Calculate the base helix angle ψ_b

$$\psi_b = \tan^{-1}(\tan(\psi_s)\ \cos(\phi_s)) = \sin^{-1}(\sin(\psi_s)\ \cos(\phi_c))$$

For spur gears $\psi_b = 0$

14. Calculate the helical factor C_ψ

a. For spurs and conventional helical gears with

$m_F = 0$ or > 1.0

$C_\psi = 1.0$

b. For LCR helical gears with $0 < m_F \leq 1.0$

$$C_\psi = [1.0 - m_F + C_{xh}\ Z\ m_F^2 / (C_x\ F\ \sin(\psi_b))]^{0.5}$$

15. Calculate the curvature factor C_c

$$C_c = 0.5\ N_G \cos(\phi_t) \sin(\phi_t) / (N_P + N_G)$$

16. Calculate the minimum length of the lines of contact L_{min}

a. For spur gears $L_{min} = F$

b. For LCR helical gears, the load sharing ratio is considered in the C_{xh} factor, and hence L_{min} is considered to be equal to F

c. For conventional helical gears the following calculations are necesary:

Calculate the transverse base pitch p_b

$$p_b = \pi\ m_n \sec(\psi_s) \cos(\phi_s)$$

Calculate the transverse contact ratio m_P

$$m_P = Z / p_b$$

Calculate the limiting number of lines of contact n

$$n = INT\ (m_P + m_F)$$

Calculate the overlap coefficient C_o

$$C_o = n - m_F$$

Note:

If $m_F > n$ then $C_o = 0$

$$L_{min} = \frac{p_b\ [m_P n - C_o - INT(m_P)F(m_P) - INT(C_o)\{INT(m_P) + F(C_o) - 1\}]}{\sin(\psi_b)}$$

where INT(X) is the truncated integer portion of X and F(X) = X - INT(X)

17. Calculate the load sharing ratio m_N

$$m_N = F / L_{min}$$

Note: For spur and LCR helical gears $m_N = 1.0$

18. Calculate the pitting resistance geometry factor I

$$I = C_x\ C_\psi^2\ C_c / m_N$$

19. Calculate the helix angle at the operating pitch diameter ψ

$$\psi = \tan^{-1}[2\ r\ \sin(\psi_s)/(N_P\ m_n)]$$

Note: For spur gears, $\psi = 0$

20. Calculate the operating normal pressure angle ϕ_n

$$\phi_n = \tan^{-1}[\tan(\phi_t)\ \cos(\psi)]$$

Note: For spur gears, $\phi_n = \phi_t$

21. Calculate the virtual spur gear dimensions

$$N_{eP} = N_P / \cos^3(\psi)$$

$$N_{eG} = N_{eP}\ N_G / N_P$$

$$r_{oe} = m_n(0.5\ N_{eP} + x_P) + h_b - \Delta r_o$$

$$R_{oe} = m_n(0.5\ N_{eG} + x_G) + h_b - \Delta R_o$$

Note:

For spur gears, $N_{eP} = N_P$; $N_{eG} = N_G$; $r_{oe} = r_o$; $R_{oe} = R_o$

22. Calculate the load angle ϕ_L

a.

$$\Omega_P = \cos^{-1}(0.5\ N_{eP}\ m_n \cos(\phi_c)/r_{oe})$$

$$\Omega_G = \cos^{-1}(0.5\ N_{eG}\ m_n \cos(\phi_c)/R_{oe})$$

b.

$$\phi_{LNP} = \tan(\Omega_P) - (0.5\pi + 2\ x_P \tan(\phi_c) - B_{NP}/m_n)/N_{eP} - \mathrm{inv}(\phi_c)$$

$$\phi_{LNG} = \tan(\Omega_G) - (0.5\pi + 2\ x_G \tan(\phi_c) - B_{NG}/m_n)/N_{eG} - \mathrm{inv}(\phi_c)$$

c. For inaccurate spur gears and conventional helical gears

$$\phi_{LP} = \phi_{LNP}$$

$$\phi_{LG} = \phi_{LNG}$$

d. For accurate spur gears

$$\phi_{LP} = \phi_{LNP} - 2(Z/m_n - \pi \cos(\phi_c))/(N_P \cos(\phi_c))$$

$$\phi_{LG} = \phi_{LNG} - 2(Z/m_n - \pi \cos(\phi_c))/(N_G \cos(\phi_c))$$

e. For LCR helicals the following calculations are necessary. Calculate the equivalent base radii r_{be} and R_{be}

$$r_{be} = 0.5\ N_{eP}\ m_n \cos(\phi_c)$$

$$R_{be} = r_{be}\ N_G\ /\ N_P$$

Calculate the equivalent operating pitch radii r_e and R_e

$$r_e = r / \cos^2(\psi)$$

$$R_e = r_e N_G / N_P$$

Calculate the equivalent length of the line of action in the normal plane Z_{ne}

$$Z_{ne} = [R_{oe}^2 - R_{be}^2]^{0.5} - [R_e^2 - R_{be}^2]^{0.5} + [r_{oe}^2 - r_{be}^2]^{0.5} - [r_e^2 - r_{be}^2]^{0.5}$$

$$\phi_{LP} = \phi_{LNP} - 2(Z_{ne} / m_n - \pi \cos(\phi_c))/(N_{eP} \cos(\phi_c))$$

$$\phi_{LG} = \phi_{LNG} - 2(Z_{ne} / m_n - \pi \cos(\phi_c))/(N_{eG} \cos(\phi_c))$$

For inaccurate spur gears and conventional helical gears the load angle is calculated at the tip. For accurate spur gears and LCR helicals the load angle is calculated at the highest point of single tooth contact (HPSTC).

If the gears are manufactured using an ISO 53 cutter, and have addendum modifications within the conventional limits of ISO/TR4467, approximate values of t_e and h_e can be obtained from charts in Section 2.

If exact values of t_e and h_e are required, or the gears are manufactured using an alternative cutter to ISO 53, values of t_e and h_e are calculated from steps 23 to 27 inclusive.

23. Set the initial value of λ

For accurate spur gears and LCR helical gears $\lambda = \phi_c$

For inaccurate spur gears and conventional helical gears $\lambda = 0.75\ \phi_c$

24. Calculate the constants K_1, K_2 and K_3

$$K_1 = (h_a - r_T)/m_n - x + 0.5\ B_N\ /\ (m_n \tan(\phi_c))$$

If $K_1 = 0.0$, calculate K_4 which then equals K_7.

$$K_2 = 0.25\ \pi + [h_a \tan(\phi_c) + r_T\ (\sec(\phi_c) - \tan(\phi_c))]/m_n$$

$$K_3 = 0.5\ N_e/K_1$$

25. Calculate the variables K_4, K_5, K_6, K_7, K_8 and K_9

$$K_4 = 0.5\ \pi - \lambda - 2K_2/N_e$$

$$K_5 = K_3\ K_4/(K_3 + 1.0)$$

$$K_6 = K_5 - (\tan(K_5) + K_5\ K_3 - K_4\ K_3)/(\sec^2(K_5) + K_3)$$

$$K_7 = K_6 - (\tan(K_6) + K_6\ K_3 - K_4\ K_3)/(\sec^2(K_6) + K_3)$$

$$K_8 = 0.5\ \pi - \lambda - K_7$$

$$K_9 = r_T/m_n + K_1 \sec(K_7)$$

26. Calculate the inscribed parabola dimensions t_e and h_e

$$t_e = m_n\ [N_e \sin K_8 - 2\ K_9 \cos(\lambda)]$$

$$h_e = m_n\{0.5\ N_e\ [\cos(\phi_c) \sec(\phi_L) - \cos(K_8)] + K_9 \sin(\lambda)\}$$

27. Calculate the revised value for λ

$$\lambda' = \tan^{-1}(0.25\ t_e/h_e)$$

If the new value of λ' varies from the original value of λ by more than 0.05 radians, re-enter the calculation at step 25 with the new value of λ' substituted for the old value of λ. If the variation is less than 0.05 radians proceed to step 28.

28. Calculate the helical factor C_h

a. For spur and LCR helical gears $C_h = 1.0$

b. For conventional helical gears with $m_F > 1$

$$\omega = \{\tan^{-1}[\tan(\psi)\ \sin(\phi_n)]\}1.8/\pi$$

$$C_h = 1/[1.0 - (\omega - \omega^2)^{0.5}]$$

Note:
If full buttressing exists, the value of C_h may be increased by 10 percent. Refer to AGMA 218.01 Clause 6.3.2.3

29. Calculate the helix angle factor K_ψ

a. For spur and LCR helical gears with $m_F \leq 1.0$

$$K_\psi = 1.0$$

b. For conventional helical gears with $m_F > 1.0$

$$K_\psi = \cos(\psi)\ \cos(\psi_s)$$

30. Calculate the tooth form factor Y

$$Y = K_\psi\ t_e^2\ \cos(\phi_n)/\{m_n\ \cos(\phi_L)[6h_e/C_h - t_e\ \tan(\phi_L)]\}$$

31. To obtain the stress correction factor K_f, the following calculations are necessary:

Calculate the operating dedenda b_P and b_G

$$b_P = r + h_a - m_n[x_P + 0.5\ N_P\ \sec(\psi_s)]$$

$$b_G = R + h_a - m_n[x_G + 0.5\ N_G\ \sec(\psi_s)]$$

Calculate the minimum root fillet radii r_{fP} and r_{fG}

$$r_{fP} = (b_P - r_T)^2/(0.5\ m_n\ N_{eP} + b_P - r_T) + r_T$$

$$r_{fG} = (b_G - r_T)^2/(0.5\ m_n\ N_{eG} + b_G - r_T) + r_T$$

Calculate the Dolan and Broghamer (12)factors H, L and M

$$H = 0.34 - 1.44\ \phi_n/\pi$$

$$L = H - 0.03$$

$$M = 0.25 + 1.80\ \phi_n/\pi$$

$$K_f = H + [t_e\ /\ r_f]^L\ [t_e\ /\ h_e]^M$$

Note:

For gears not manufactured by a hob or racked shaped cutter, refer to AGMA 218.01 Clause 6.3.2.1 for the calculation of r_f.

32. Calculate the strength geometry factor J

$$J = Y\ C_\psi\ /\ (K_f\ m_N)$$

These procedural steps have been converted to a FORTRAN 77 program which is listed in Appendix A.

SECTION 2

CHARTING THE LAMBDA METHOD

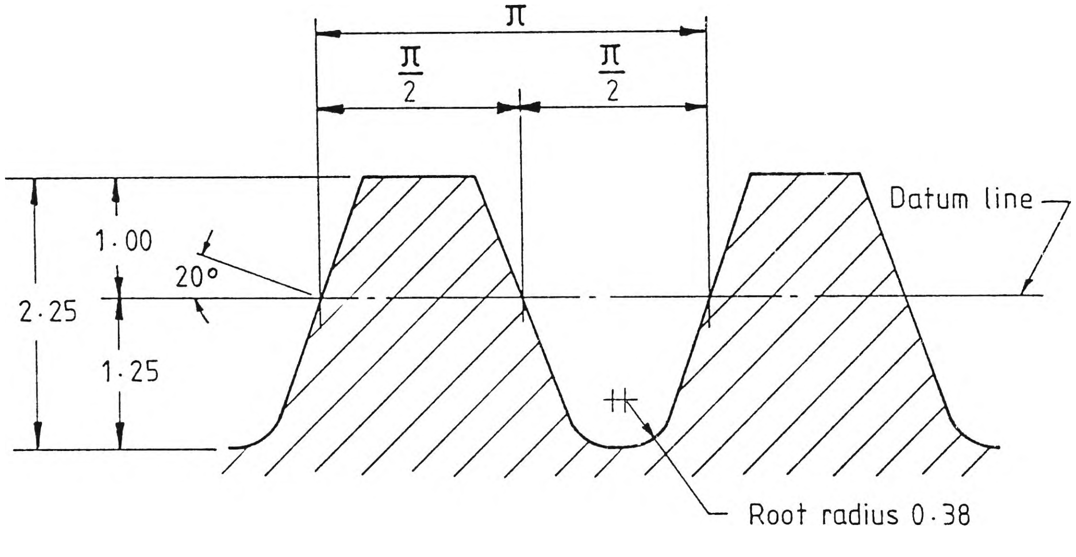


FIGURE 2.1 - THE ISO 53 GENERATING RACK

2.1 Introduction

When considering the power capacity of a gear pair based on strength, the majority of national gear codes including the American Standard AGMA 218.01 (4) require the determination of the height and width of the Lewis equal strength parabola. From these two dimensions, the geometry factor for bending strength can be determined.

If, for the particular design being considered, design charts or tabulated data are not available, then the calculation of the geometry factor for bending strength may necessitate a layout of the tooth profile. For the AGMA system, details of the tooth layout procedure are given in AGMA 226.01 (2).

The accuracy of the tooth layout procedure is very dependent on the skill of the draftsman. Also, it is unwieldly in practice when very large radii must be drawn. For a particular cutter, six charts can be prepared which eliminate the need to draw a tooth profile.

The charts in this Section give the approximate height and width of the Lewis equal strength parabola for gear teeth cut with an ISO 53(6) cutter (Figure 2.1), and have been produced utilising the Lambda Method described in Section 1 in combination with various subroutines.

The charts are restricted to addendum modification coefficients within the conventional limits of ISO/TR 4467(7). Each chart has been produced using unit normal module and incorporates backlash of 0.024mm per unit module applied to both the pinion and the wheel in the normal plane.

The charts may be categorised as those applicable to conventional helical gears and those applicable to spur gears and low contact ratio helical gears, based on the criteria of AGMA 218.01.

Examples of the use of the design charts are included and the results obtained compared with those calculated using the Lambda Method.

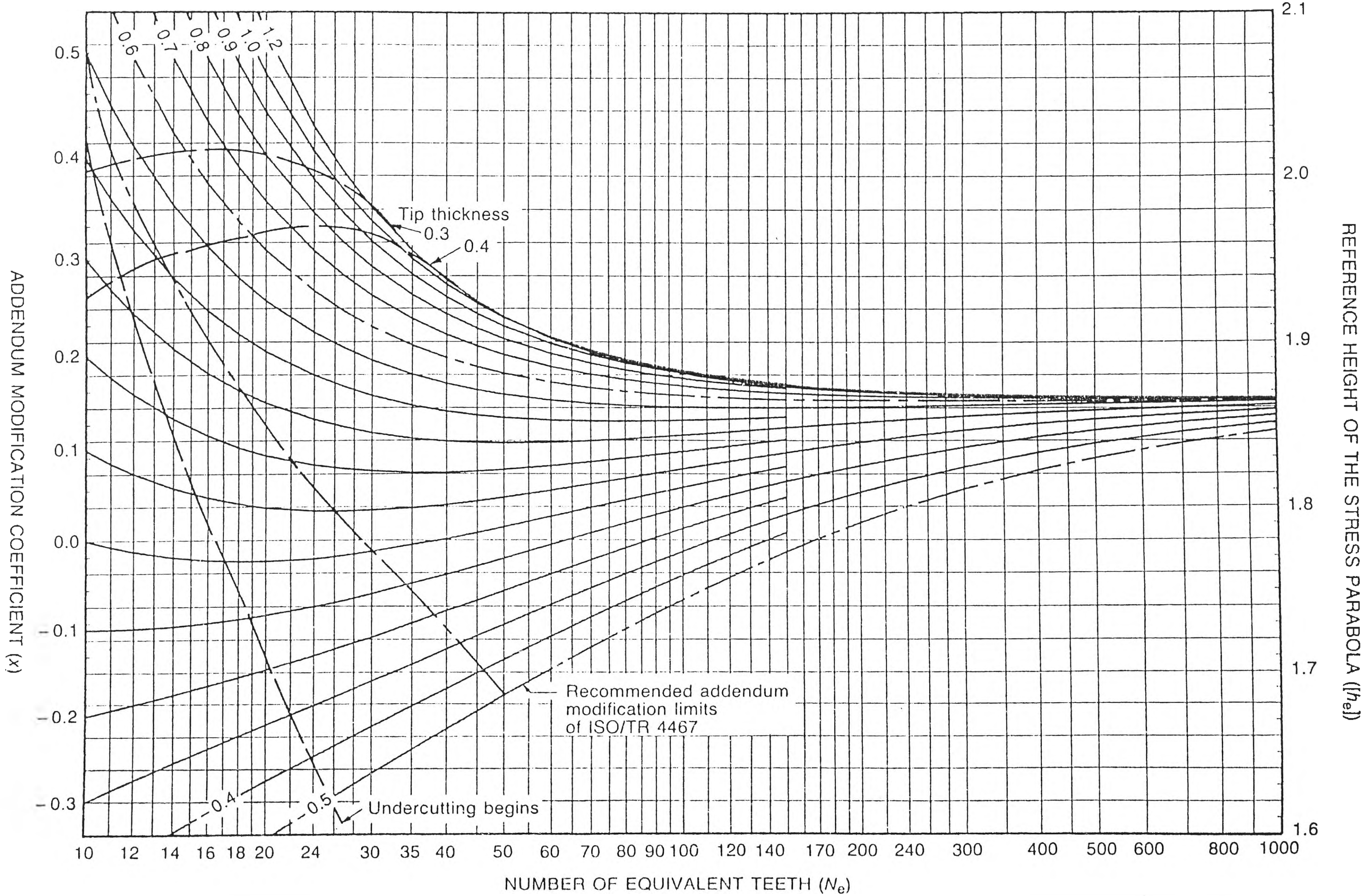


FIG. 2·2 ISO 53 REFERENCE PARABOLA HEIGHTS FOR CONVENTIONAL HELICAL GEARS

2.2 Conventional Helical Gears

Conventional helical gears by definition have a face contact ratio greater than 1.0. Figures 2.2 and 2.3 give reference parabola heights and widths respectively for conventional helical gears. The word "reference" is used to indicate that the parabola dimensions plotted in Figures 2.2 and 2.3 are for gears of unit normal module. The actual parabola dimensions are obtained by multiplying the "reference" dimensions by the actual normal module of the gear being considered.

Figures 2.2 and 2.3 have been obtained by using computer graphics. Unlike existing methods involving the use of graphs for determining geometry factors for bending strength, these two design charts can be used for any helix angle and addendum modification co-efficient within the conventional limits of ISO/TR 4467.

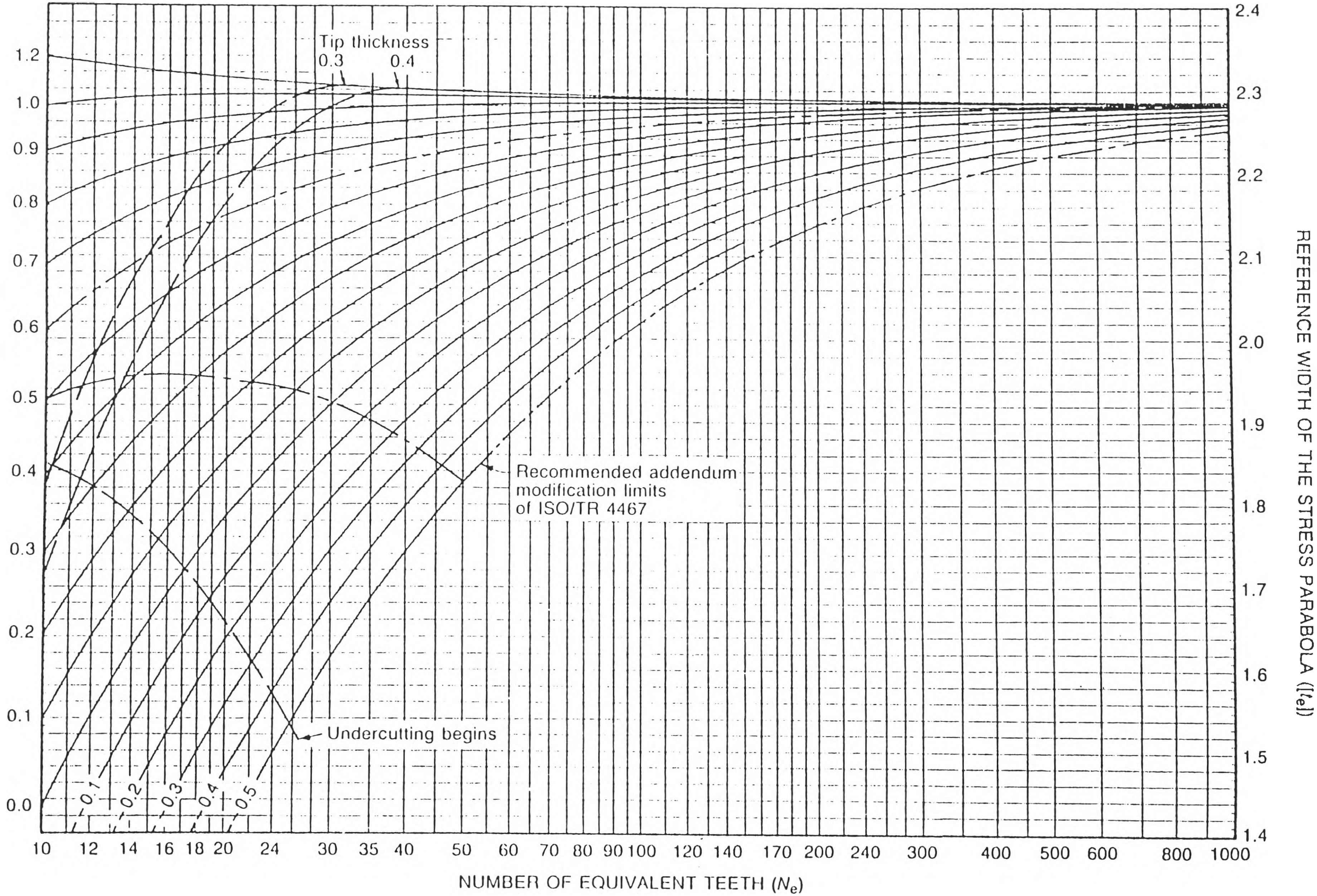


FIG. 2.3 ISO 53 REFERENCE PARABOLA WIDTHS FOR CONVENTIONAL HELICAL GEARS

2.3 Spur Gears and LCR Helical Gears

Accurate spur gears are assumed to develop the most critical stress when the load is applied at the highest point of the tooth where a single pair of teeth is carrying all the load. Helical gears with a face contact ratio of less than or equal to 1.0 are classified as low contact ratio helical gears and are assumed to be loaded in the same way as spur gears.

Figures 2.4 and 2.5 give reference parabola heights and multipliers respectively for spur gears and low contact ratio helical gears. Similarly, Figures 2.6 and 2.7 give reference parabola widths and multipliers respectively for spur gears and low contact ratio helical gears.

For each of these two types of gears, the word "reference" is used to indicate two things. Firstly, as in the case of conventional helical gears, the parabola dimensions plotted are for gears of unit normal module. Secondly, Figures 2.4 and 2.6 are applicable to a unique addendum modification coefficient and hence other addendum modification coefficients must be referenced back to this datum. The actual parabola dimensions are obtained by multiplying the "reference" dimensions by the actual normal module of the gear being considered and the appropriate multiplier.

Previous design charts, such as Figure 2 of AGMA 226.01, highlight the difficulty of plotting geometry factors for bending strength for spur gears. This difficulty arises from the variation in load angle as a function of the number of teeth in the mating gear and is further compounded by variations in addendum modification coefficients. These problems are resolved by the use of the following charts as they encompass all possible load angles and a large spectrum of addendum modification coefficients.

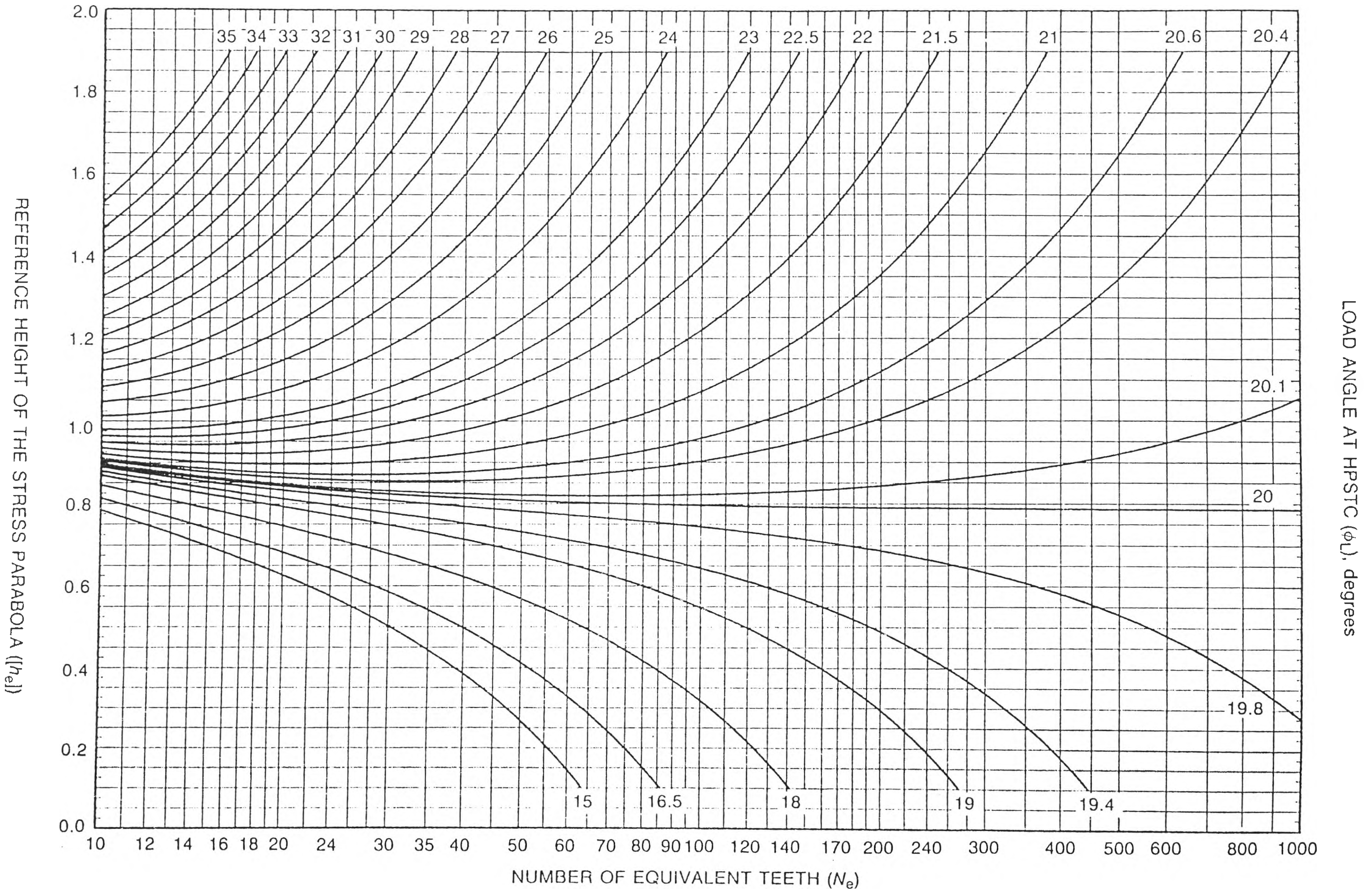


FIG. 2·4 ISO 53 REFERENCE PARABOLA HEIGHTS FOR SPUR AND LCR HELICAL GEARS

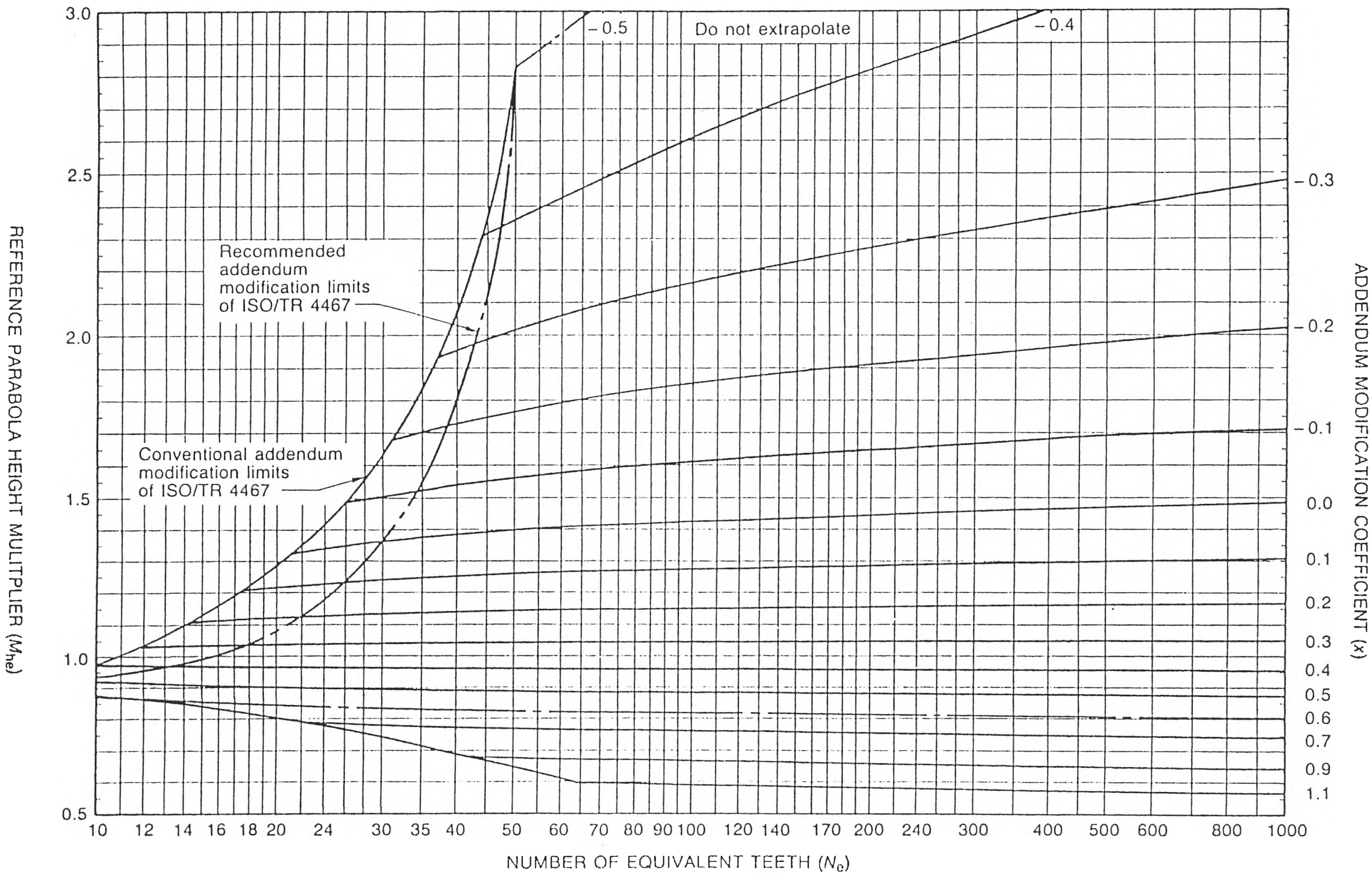


FIG. 2·5 ISO 53 REFERENCE PARABOLA HEIGHT MULTIPLIERS FOR SPUR AND LCR HELICAL GEARS

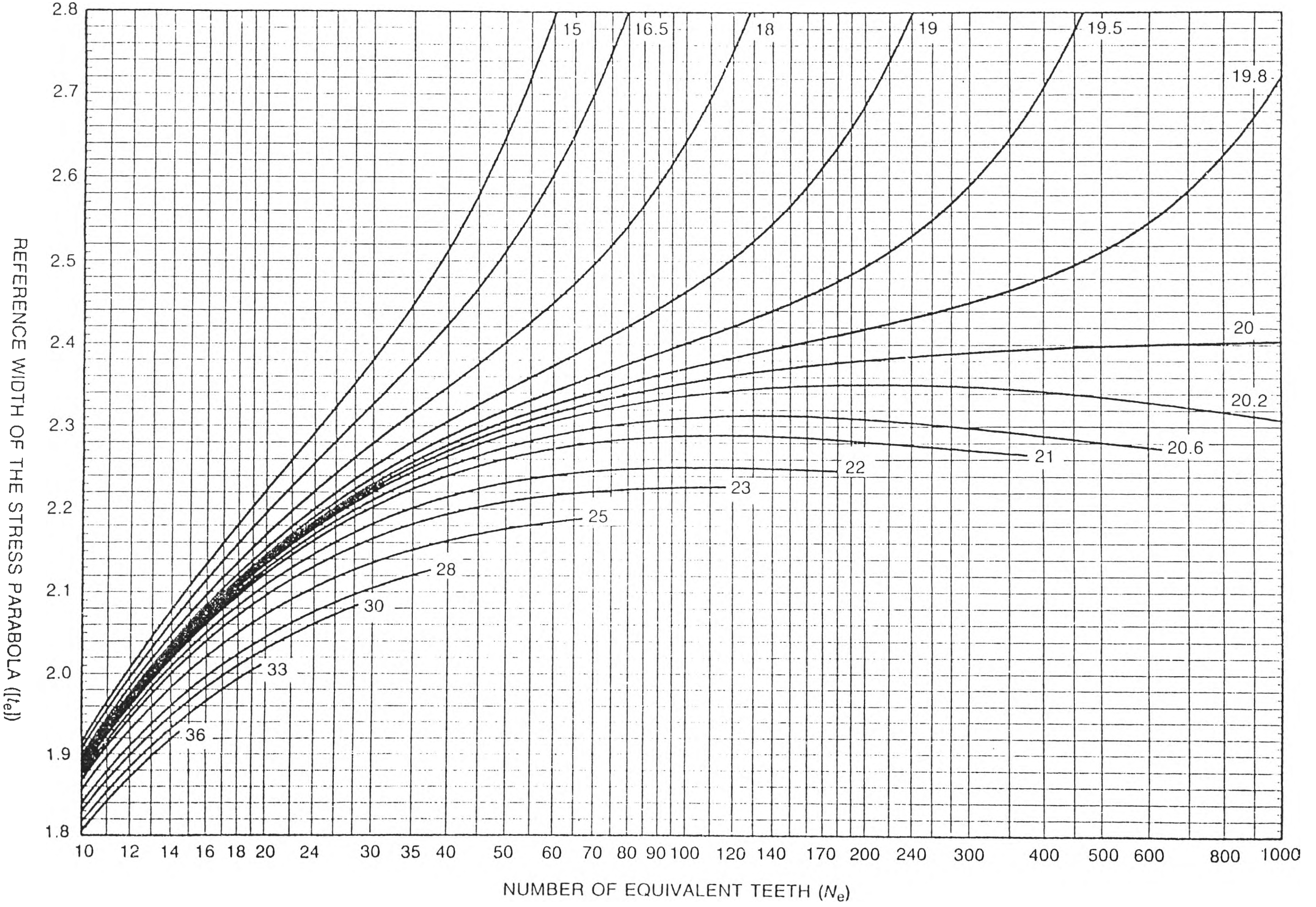


FIG. 2.6 ISO 53 REFERENCE PARABOLA WIDTHS FOR SPUR AND LCR HELICAL GEARS

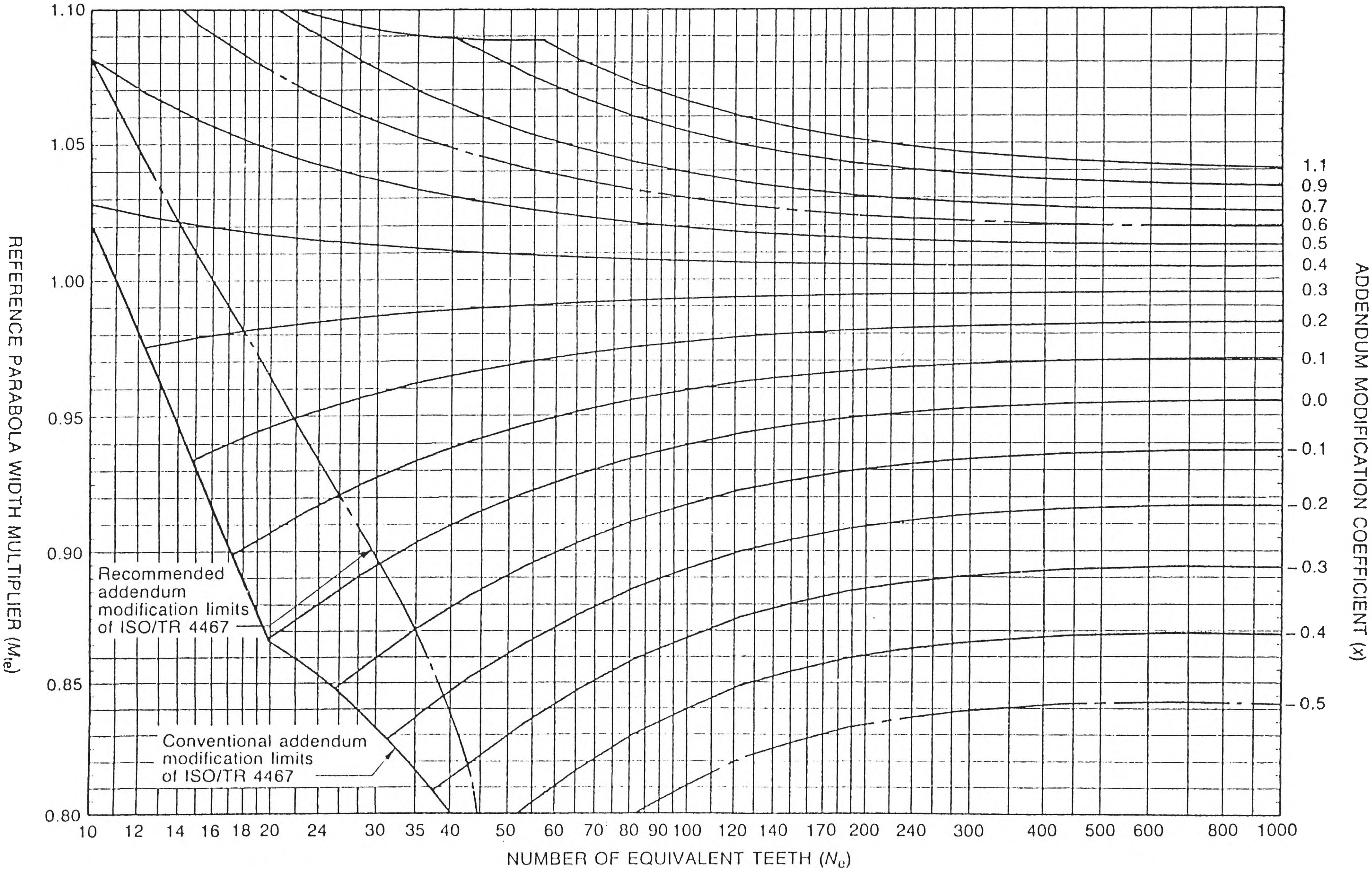


FIG. 2·7 ISO 53 REFERENCE PARABOLA WIDTH MULTIPLIERS FOR SPUR AND LCR HELICAL GEARS

2.4 Examples Of The Use Of The Design Charts

The first example is that of an accurate external spur gear set manufactured with an ISO 53 cutter.

Description	Symbol	Pinion	Wheel
Number of teeth	N_P, N_G	19	168
Add.mod.coeff	x_P, x_G	0.4000	0.6389
Backlash, mm	B_{NP}, B_{NG}	0.3668	0.9628
Topping, mm	Δr_o, ΔR_o	0.0	2.0
Module, mm	m_n	20.0	20.0
Facewidth, mm	F	147	147

For an ISO 53 cutter, $\phi_c = 20^o$ and $h_b = m_n$

The following steps calculate the pinion stress parabola dimensions. From the equations given previously in Section 1.4, or in ASME Paper 84-DET-182 (13), the load angle for the pinion is calculated to be 24.3^o.

$$[h_{eP}] = 1.095 \text{ (Figure 2.4)}$$

$$M_{heP} = 0.965 \text{ (Figure 2.5)}$$

$$h_{eP} = 1.095 \times 0.965 \times 20 = 21.134\text{mm}$$

$$[t_{eP}] = 2.078 \text{ (Figure 2.6)}$$

$$M_{teP} = 1.018 \text{ (Figure 2.7)}$$

$$t_{eP} = 2.078 \times 1.018 \times 20 = 42.308\text{mm}$$

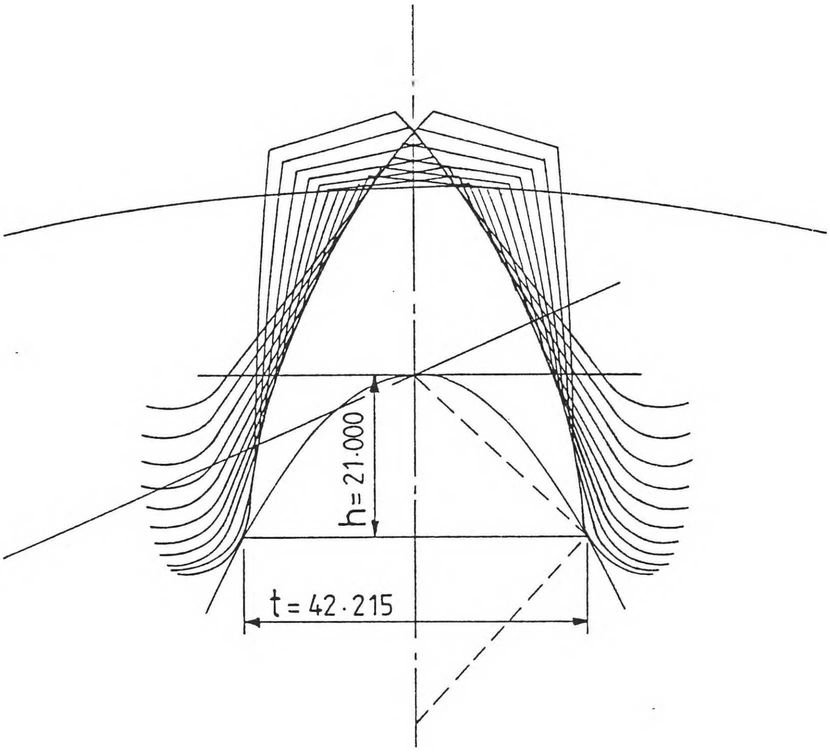


FIG. 2.8 SPUR PINION PARABOLA

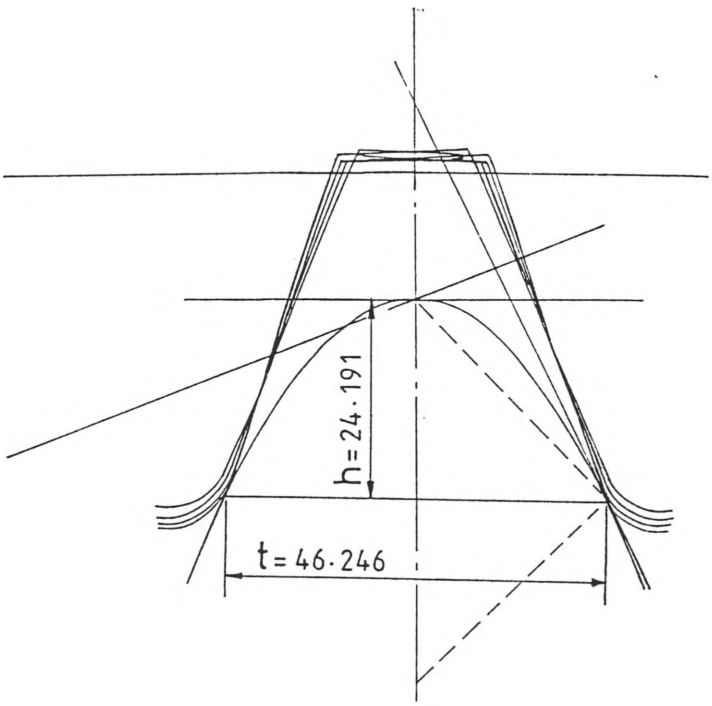


FIG. 2.9 SPUR WHEEL PARABOLA

Similarly, from the equations given previously, the load angle for the wheel is calculated to be 21.3^{o}.

$$[h_{eG}] = 1.450 \text{ (Figure 2.4)}$$

$$M_{heG} = 0.795 \text{ (Figure 2.5)}$$

$$h_{eG} = 1.450 \times 0.795 \times 20 = 23.055\text{mm}$$

$$[t_{eG}] = 2.275 \text{ (Figure 2.6)}$$

$$M_{teG} = 1.028 \text{ (Figure 2.7)}$$

$$t_{eG} = 2.275 \times 1.028 \times 20 = 46.774\text{mm}$$

The dimensions calculated using the Lambda Method and superimposed on a CAD plot are shown in Figures 2.8 and 2.9.

The second example is a set of conventional external helical gears cut with an ISO 53 cutter.

Description	Symbol	Pinion	Wheel
Number of teeth	N_P, N_G	23	84
Add. mod.coeff	x_P, x_G	0.2900	0.1342
Helix angle, DMS	ψ_s	$16^{o}37'58''$	$16^{o}37'58''$
Backlash, mm	B_{NP}, B_{NG}	0.1815	0.2791
Topping, mm	Δr_o, ΔR_o	0.0	0.0
Normal module, mm	m_n	8.0	8.0
Facewidth, mm	F	180	180

For an ISO 53 cutter, $\phi_c = 20^{o}$ and $h_b = m_n$

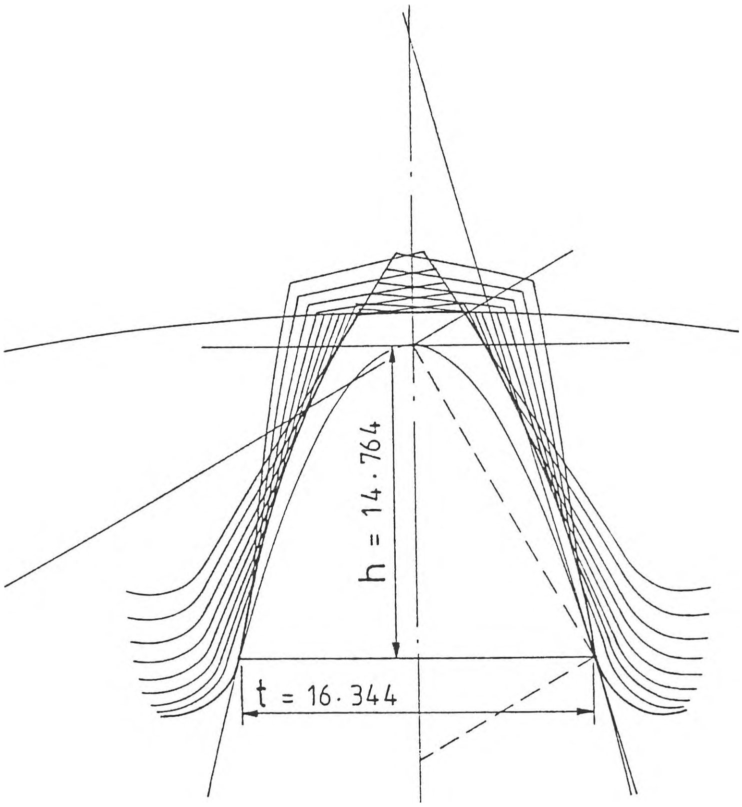


FIG. 2·10 CONVENTIONAL HELICAL PINION PARABOLA

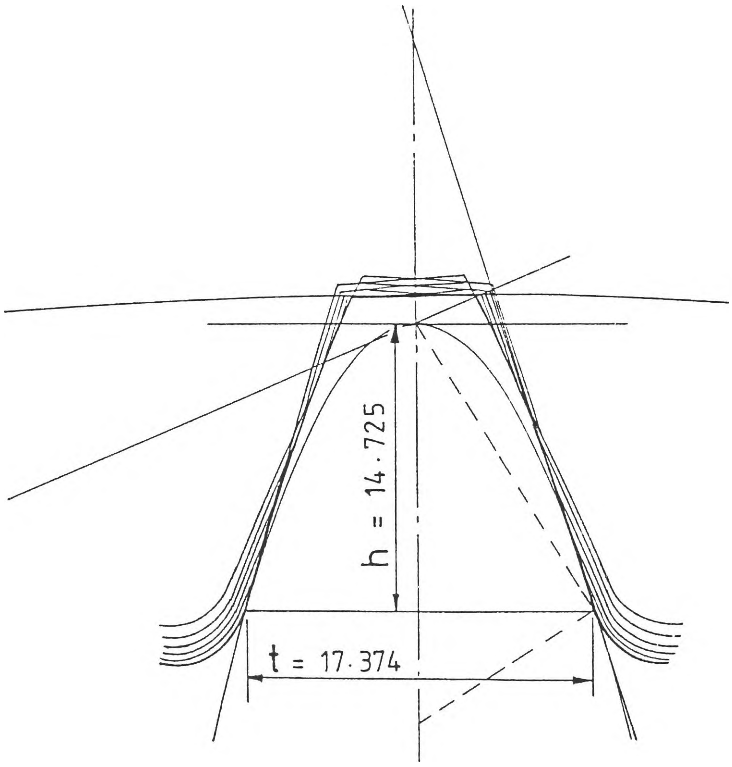


FIG. 2·11 CONVENTIONAL HELICAL WHEEL PARABOLA

From the equations given previously in Section 1.9, the equivalent number of pinion teeth is calculated to be 26.2.

Note: AGMA 218.01 calculates the equivalent number of teeth based on the helix angle at the operating pitch diameter as opposed to earlier publications which were based on the helix angle at the standard pitch diameter.

$$[h_{eP}] = 1.850 \text{ (Figure 2.2)}$$

$$h_{eP} = 1.850 \times 8 = 14.800\text{mm}$$

$$[t_{eP}] = 2.035 \text{ (Figure 2.3)}$$

$$t_{eP} = 2.035 \times 8 = 16.280\text{mm}$$

From the equations given previously, the equivalent number of wheel teeth is calculated to be 95.7.

$$[h_{eG}] = 1.830 \text{ (Figure 2.2)}$$

$$h_{eG} = 1.830 \times 8 = 14.640 \text{ mm}$$

$$[t_{eG}] = 2.175 \text{ (Figure 2.3)}$$

$$t_{eG} = 2.175 \times 8 = 17.400 \text{ mm}$$

The dimensions calculated using the Lambda Method and superimposed on a CAD plot are shown in Figures 2.10 and 2.11.

The third example is that of a LCR helical gear set manufactured with an ISO 53 cutter.

The conventional helical gear set analysed previously is accurately hobbed to form a double helical gear, the central tool clearance gap between each helix being 20mm, thus reducing the net facewidth to 80mm, such that $m_F = 0.9111$.

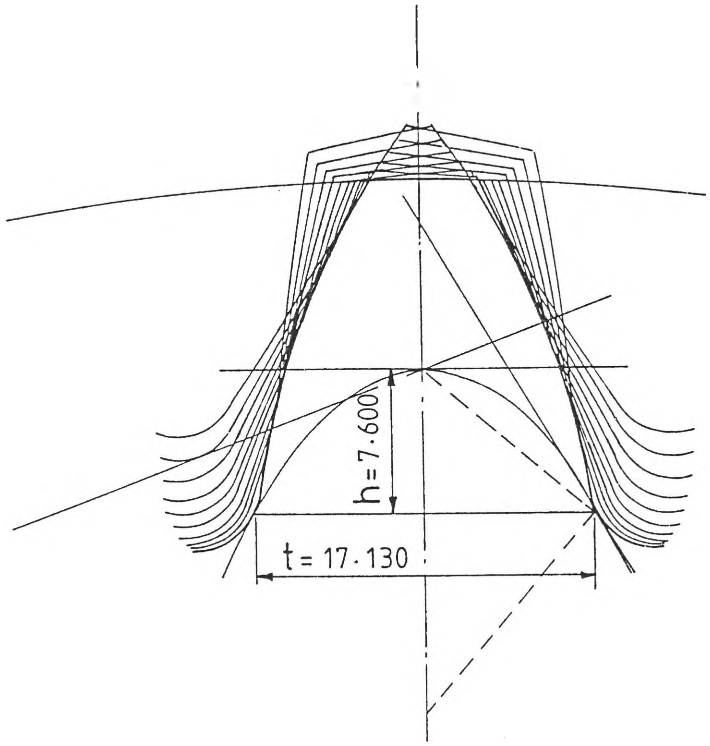


FIG. 2.12 LCR HELICAL PINION PARABOLA

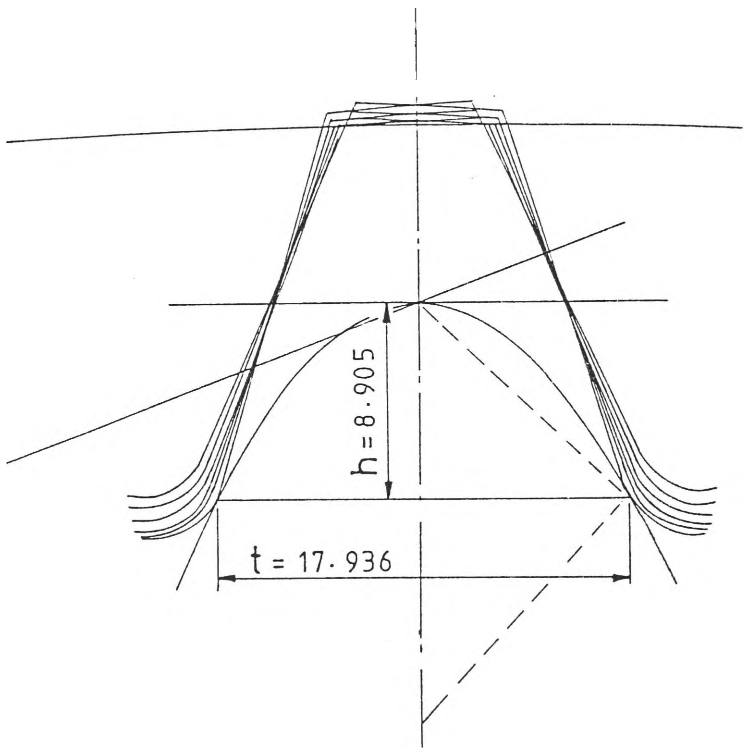


FIG. 2.13 LCR HELICAL WHEEL PARABOLA

From the equations given previously, the load angle for the pinion is calculated to be 21.1^{o} while the equivalent number of pinion teeth is 26.2.

$[h_{eP}] = 0.913$ (Figure 2.4)

$M_{heP} = 1.055$ (Figure 2.5)

$h_{eP} = 0.913 \times 1.055 \times 8 = 7.706$ mm

$[t_{eP}] = 2.175$ (Figure 2.6)

$M_{teP} = 0.983$ (Figure 2.7)

$t_{eP} = 2.175 \times 0.983 \times 8 = 17.104$mm

From the equations given previously, the load angle for the wheel is calculated to be 20.5^{o} while the equivalent number of wheel teeth is 95.7.

$[h_{eG}] = 0.925$ (Figure 2.4)

$M_{heG} = 1.240$ (Figure 2.5)

$h_{eG} = 0.925 \times 1.240 \times 8 = 9.176$mm

$[t_{eG}] = 2.319$ (Figure 2.6)

$M_{teG} = 0.965$ (Figure 2.7)

$t_{eG} = 2.319 \times 0.965 \times 8 = 17.903$mm

The dimensions calculated using the Lambda Method and superimposed on a CAD plot are shown in Figures 2.12 and 2.13.

SECTION 3

CHARTING BENDING STRENGTH GEOMETRY FACTORS

3.1 Introduction

There are many gear applications which do not necessitate an optimisation of the design and as such a simple method to rate the gears would be advantageous.

One of the most difficult variables to calculate in the rating of gears is that of the bending strength geometry factor, J. Whilst AGMA 218.01(4) Clause 6.3 details the procedure for the calculation of J, and Figures B1 to B10 inclusive have charted values of J for several different cutters, there are no charts for the geometry factor for bending strength for gears operating at extended centre distances. Further, AGMA 218.01 has, understandably, no charts for an ISO 53(6) cutter.

The Standards Assocation of Australia realised the need to have J factor charts for an ISO 53 cutter, when AGMA 218.01 was adopted as the Australian Standard, in combination with the recommendation to use an ISO 53 cutter as the preferred standard cutter.

In response to a request from the Standard Association of Australia, eighteen charts were produced for AS 2938(5), two of which are highlighted in this Section.

3.2 Rationalisation of Variables

Clause 6.3 of AGMA 218.01 gives the equations which are necessary for the calculation of the geometry factor for bending strength J. An examination of these equations reveals that the geometry factor for bending strength is a function of the number of teeth in the pinion and wheel, the normal profile angle, addendum, dedendum and tip radius of the standard counterpart rack, the normal module, the helix angle at the standard pitch circle diameter, the addendum modification coefficients of the pinion and wheel with zero backlash, the net facewidth of the narrowest member, the radial tooth truncation and the backlash applied in the normal plane. This relationship may be expressed as

$$J = f(N_P, N_G, \phi_c, h_a, h_b, r_T, m_n, \psi_s, x_P, x_G, F, \Delta r_o, B_N) \quad (3.1)$$

The relationship is further compounded, when for inaccurate spur gears and conventional helical gears the load is calculated at the tooth tip, whilst for accurate spur gears and LCR helicals, the load angle is calculated at the highest point of single tooth contact (HPSTC).

To enable charts to be developed for the geometry factor for bending strength, limits were placed on several of the variables in equation (3.1). This rationalisation of the variables is summarised as follows:

i. It can be shown that J is independent of the normal module, m_n. Thus m_n may be eliminated from equation (3.1).

ii. Only full depth teeth (ie, $\Delta r_o = 0.0$; $\Delta R_o = 0.0$) manufactured with an ISO 53 cutter are considered. Hence for the series of charts in AS 2938, $\phi_c = 20^o$, $h_a = 1.25$ mm, $h_b = 1.0$ mm and $r_T = 0.38$ mm.

iii. For conventional helical gears, the approximation given by equation (6.27) of AGMA 218.01 for the load sharing ratio, m_N, for a face contact ratio, $m_F > 2.0$, has been adopted. This approximation of $m_N = p_N/(0.95\ Z)$, eliminates facewidth from the calculation of the load sharing ratio (refer to Section 1.8) and thus from the geometry factor for bending strength.

iv. The addendum modification coefficients of the pinion and wheel, x_P and x_G, are set as constants for each chart.

v. The backlash in the normal plane, B_N, is set at a constant 0.024mm per unit normal module for both the pinion and wheel, to provide a total backlash of 0.048mm. This technique is in keeping with that adopted by AGMA 218.01.

vi. All gears considered are external helical gears, where N_p is never considered to be greater than N_G.

vii. All spur gears are considered to be accurate, and hence the load angle, ϕ_L, is calculated at the HPSTC.

viii. All helical gears are considered to have a face contact ratio, m_F, of 2.0 or greater, and hence the load angle, ϕ_L, is calculated at the tooth tip.

ix. Low contact ratio (LCR) helical gears are not considered, due to the need to eliminate the facewidth, F, from the calculations.

3.3 Development of the J Factor Charts

As a result of the rationalisation of the variables, then for a variable number of pinion teeth, N_P, and a fixed number of wheel teeth, N_G, equation (3.1) may be reduced to that of equation (3.2).

$$J_P = J_P(N_P, N_G, \Psi_S) \tag{3.2}$$

A normal intercept chart may be constructed from equation (3.2), but this technique has the disadvantage of requiring a large number of charts, for varying numbers of wheel teeth and addenda modifications.

An examination of the series of charts, in conjunction with those in AGMA 218.01, showed that a multiplying chart could be produced, thus limiting the number of charts to two for each addendum modification, whilst each chart had the ability to accommodate varying numbers of wheel teeth.

A statistical analysis has shown, however, that equation (3.2) may be approximated with reasonable accuracy, by,

$$J_P = J_P(N_P, N_G = N_G', \Psi_S)\ M_P(N_G, \Psi_S) \tag{3.3}$$

$$\text{where } M_P(N_G, \Psi_S) = \left[\sum_{i=1}^{i=n} \frac{J_P(N_{P(i)}, N_G, \Psi_S)}{J_P(N_{P(i)}, N_G = N_G', \Psi_S)} \right] / \ n \tag{3.4}$$

$N_{P(i)}$ = number of reference pinion teeth selected

n = number of reference increments

N_G' = reference number of wheel teeth

The equations for the charting of the wheel, can be derived by considering the wheel as the pinion in the preceding analysis.

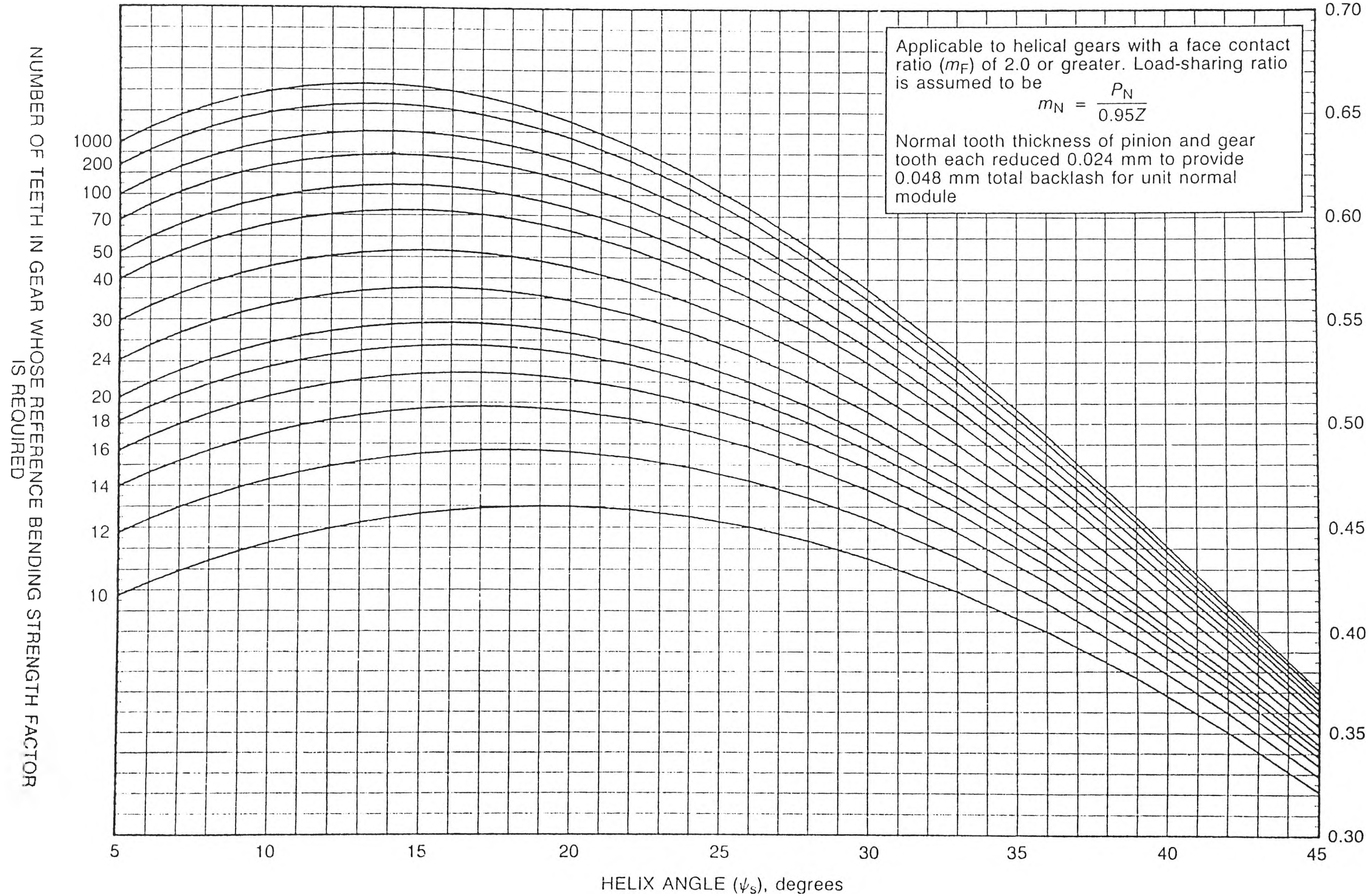


FIG. 3·1 [J] FOR CONVENTIONAL HELICAL GEARS - ISO 53 CUTTER ($x_P = 0{\cdot}2$, $x_G = 0{\cdot}2$)

The mathematical form described lends itself to the development of a dual chart system for the graphical representation of the geometry factor for bending strength for accurate spur and conventional helical gears. The first chart represents $J_P(N_P, N_G = N_G', \Psi_s)$, whilst the second chart represents $M_P(N_G, \Psi_s)$. These charts have been nominated as the reference and multiplier charts respectively.

The reference charts were developed in the form of an intercept chart using computer graphics. The reference number of gear teeth, N_G', was set as 75, being the same reference value, as that adopted by AGMA 218.01. An example of a reference chart is given in Figure 3.1.

The multiplier charts were investigated using a computer program which generated five arrays. These five arrays were; the number of teeth in the meshing gear varying from 10 to 1000, the helix angle varying from 5^o to 45^o, the geometry factor for bending strength of the meshing gear, when mated with a constant number of gear teeth whose multiplication factor was desired, and the corresponding value of the multiplication factor. The multiplier chart associated with Figure 3.1 is shown as Figure 3.2.

To illustrate how the algorithm of equation (3.4) was utilised to produce the bending strength geometry factor multiplier, M_J, the following analysis indicates the calculations involved in finding the numerical value of one co-ordinate of Figure 3.2.

Selecting a reference number of gear teeth of 75, a constant helix angle of 12^o and varying the number of reference teeth in the pinion, $N_{P(i)}$, from 10 to 40, ten values of the reference bending strength geometry factor, [J], were calculated and are tabulated in Table 3.1. The numerical values of [J] as tabulated in Table 3.1, are the ordinates which have been plotted on the 12^o abscissa of Figure 3.1. It should be noted that the calculated values of [J], assume that for the particular pair of gears being analysed, there is no tip to root fillet interference. For the 10:75 combination, this is not a valid assumption, as the wheel would require radial tooth truncation. However, this point has not been included, due to the approximate nature of the graph.

i	ψ_s	$N_{P(i)}$	N_G'	[J]
1	12	10	75	0.450
2	12	12	75	0.480
3	12	14	75	0.502
4	12	16	75	0.519
5	12	18	75	0.533
6	12	20	75	0.545
7	12	22	75	0.554
8	12	24	75	0.562
9	12	30	75	0.581
10	12	40	75	0.601

TABLE 3.1 - REFERENCE BENDING STRENGTH GEOMETRY FACTOR [J]

Consider the same set of calculations, but changing the reference number of gear teeth to 40. The new values of J are tabulated in Table 3.2.

i	ψ_s	$N_{P(i)}$	N_G	J
1	12	10	40	0.441
2	12	12	40	0.470
3	12	14	40	0.492
4	12	16	40	0.508
5	12	18	40	0.521
6	12	20	40	0.532
7	12	22	40	0.541
8	12	24	40	0.549
9	12	30	40	0.566
10	12	40	40	0.585

TABLE 3.2 - BENDING STRENGTH GEOMETRY FACTOR J

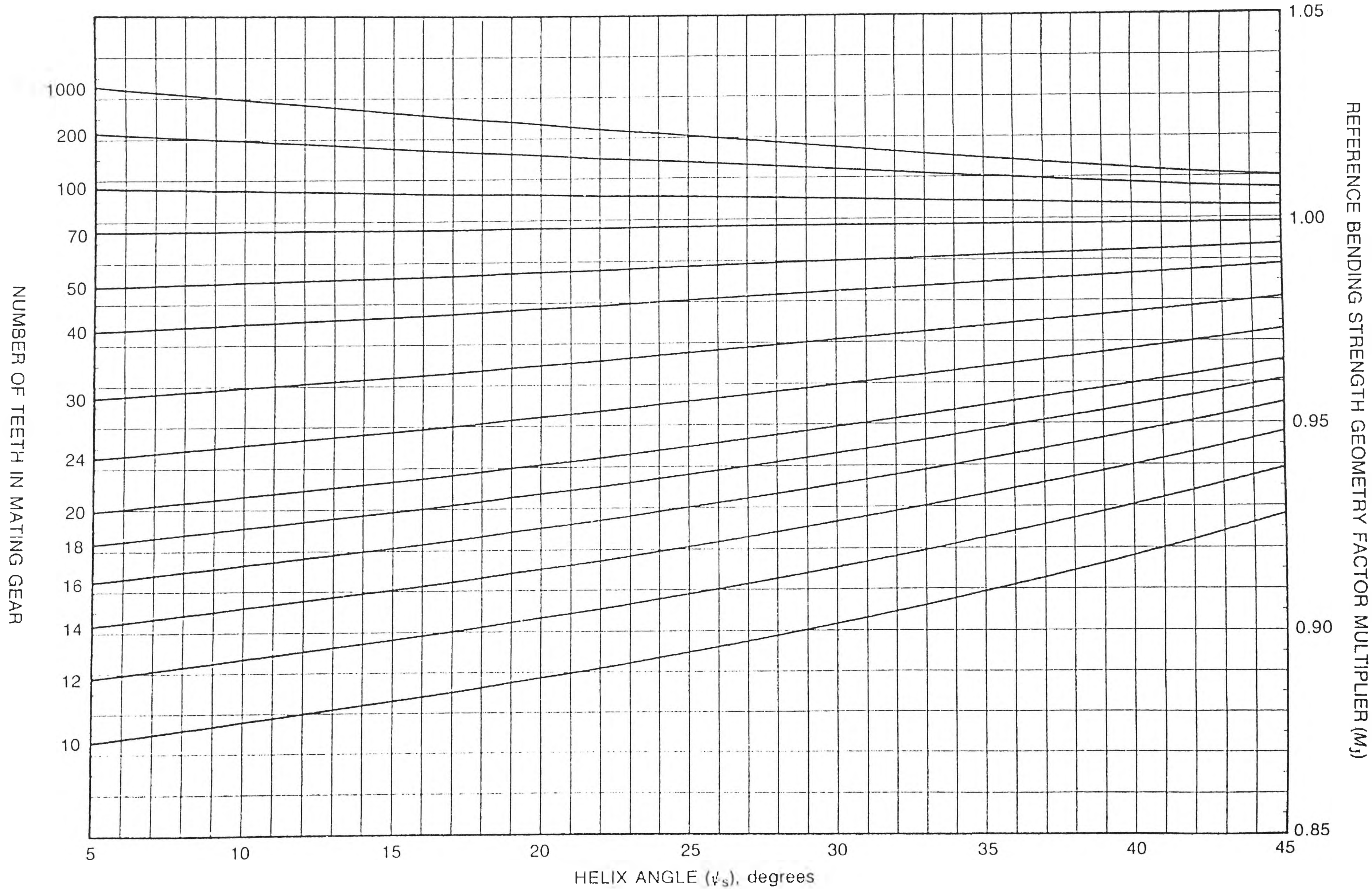


FIG. 3·2 M_J FOR CONVENTIONAL HELICAL GEARS - ISO 53 CUTTER ($x_P = 0{\cdot}2$, $x_G = 0{\cdot}2$)

The multiplying factor, $M_J = M_P(N_G, \Psi_s) = M_P(40,12)$, can now be calculated from the ratio J:[J]. The values obtained are tabulated in Table 3.3.

i	J	[J]	M_J=J/[J]
1	0.441	0.450	0.9800
2	0.470	0.480	0.9792
3	0.492	0.502	0.9801
4	0.508	0.519	0.9788
5	0.521	0.533	0.9775
6	0.532	0.545	0.9761
7	0.541	0.554	0.9765
8	0.549	0.562	0.9769
9	0.566	0.581	0.9742
10	0.585	0.601	0.9734
			Σ 9.7727

TABLE 3.3 - BENDING STRENGTH GEOMETRY MULTIPLIER M_J

From equation (3.4),

$$M_J = M_P(40,12) = \left[\sum_{i=1}^{10} \frac{J_P(N_{P(i)},40,12)}{J_P(N_{P(i)},75,12)} \right] / 10$$

$$= 9.7727/10$$

$$= 0.977$$

This co-ordinate may be found on Figure 3.2.

3.4 Examples of the Use of the Design Charts

An example is presented to demonstrate the versatility of the J factor charts. Further, the results obtained for the geometry factor for bending strength for both the pinion and the wheel are compared with those calculated using the procedural steps of Section 1.9. This comparison demonstrates the high order of accuracy which can be achieved using the design charts.

Consider the example of a set of conventional helical gears running at an extended centre distance.

Description	Pinion	Wheel
Number of teeth	40	200
Addendum modification, mm	1.6	1.6
Helix angle, degrees	30.0	30.0
Cutter	ISO 53	ISO 53
Backlash, mm	0.192	0.192
Normal module, mm	8.0	8.0
Facewidth, mm	200.0	200.0

The following steps calculate J_p and J_G using the charts

$[J_p] = 0.530$ (Figure 3.1)

$M_{JP} = 1.012$ (Figure 3.2)

$J_P = 0.530 \times 1.012 = 0.536$

$[J_G] = 0.560$ (Figure 3.1)

$M_{JG} = 0.982$ (Figure 3.2)

$J_G = 0.560 \times 0.982 = 0.550$

SECTION 4

CHARTING PITTING RESISTANCE GEOMETRY FACTORS

4.1 Introduction

AGMA 218.01 (4) presents details of a procedure to rate any pair of spur or helical gears produced with any cutter and any addendum modification. However, many gear designs use a standard cutter for which the gear designer usually requires a quick method for determining the power rating of the gears.

This can be achieved from AGMA 218.01 by the use of charts, with one notable exception, namely that of the geometry factor for pitting resistance for helical gears.

This Section presents charts for the geometry factor for pitting resistance for conventional helical gears with full depth teeth and 20° normal pressure angle. These charts complete those which are necessary to enable the conventional helical gear pair to be rated.

Clause 6.2 of AGMA 218.01 gives the equations which are necessary for the calculation of the geometry factor for pitting resistance, I. An examination of these equations reveals that the geometry factor for pitting resistance for conventional helical gears is a function of the number of teeth in the pinion and wheel, the normal profile angle of the counterpart rack, the standard normal metric module, the helix angle at the standard pitch diameter, the addendum modification co-efficient of the pinion and wheel with zero backlash, the net facewidth of the narrowest member and the standard dedendum of the counterpart rack. This relationship may be expressed as

$$I = f(N_P, N_G, \phi_c, m_n, \psi_s, x_P, x_G, F, h_b) \qquad (4.1)$$

To enable charts to be developed for the geometry factor for pitting resistance for conventional helical gears, limits were placed on several of the variables in equation (4.1). This rationalisation of the variables is summarised on the following page.

i. It can be shown that I is independent of the standard normal metric module, m_n. Thus m_n may be eliminated from equation (4.1).

ii. Only full depth teeth and a normal profile angle of 20^o are considered in the examples presented, ie, $\phi_c = 20^o$ and $h_b = 1.0$ mm. Charts for stub teeth and normal pressure angles other than 20^o have been developed.

iii. For conventional helical gears, the approximation given by equation (6.27) of AGMA 218.01 for the load sharing ratio, m_N, for a face contact ratio, $m_F > 2.0$, has been adopted. This approximation of $m_N = p_N/(0.95\ Z)$, eliminates facewidth from the calculation of the load sharing ratio and thus from the geometry factor for pitting resistance.

iv. The addendum modification coefficients of the pinion and wheel were set as constants for each chart.

v. All gears considered are external helical gears, where N_P is never considered to be greater than N_G.

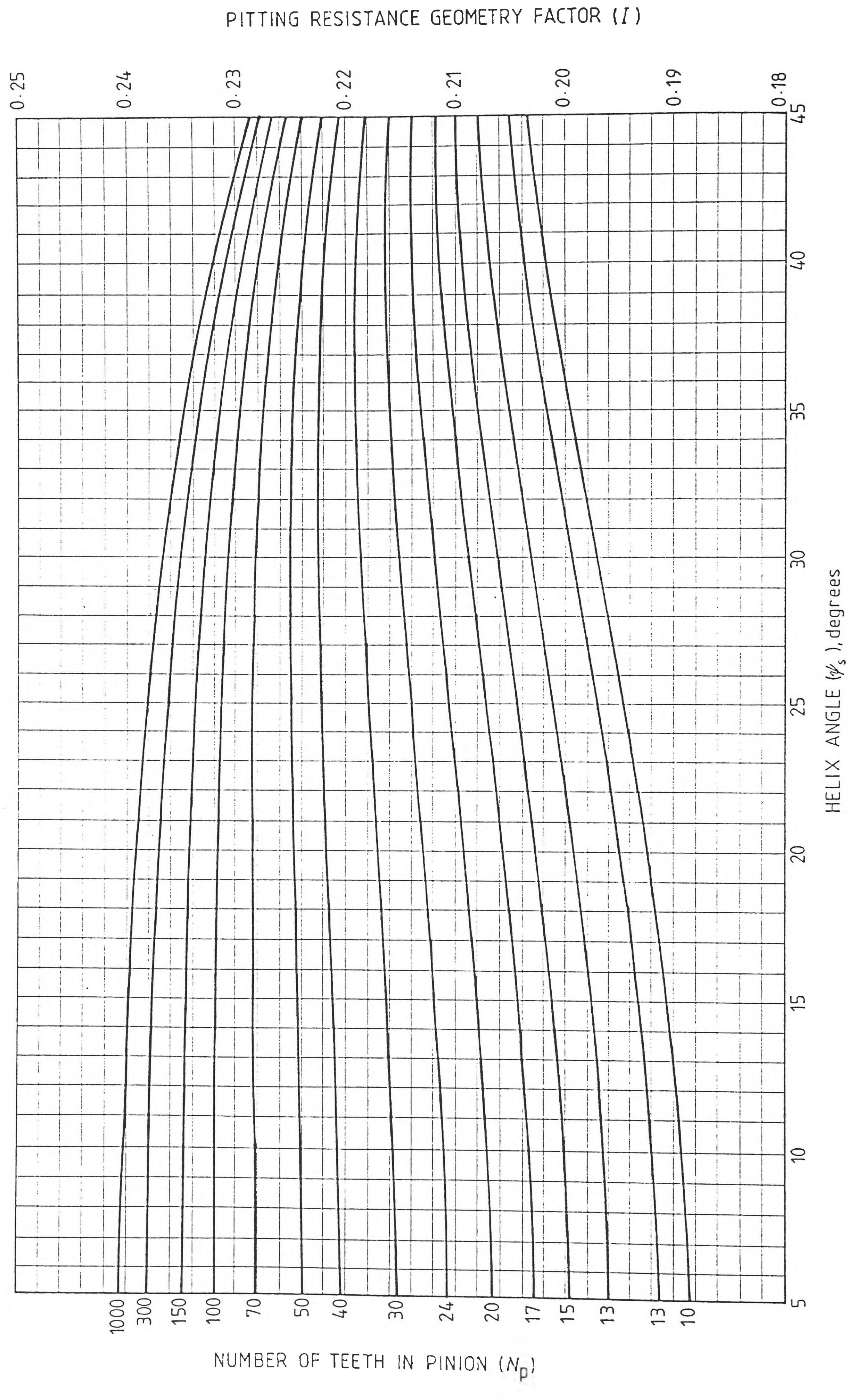


FIG. 4.1 I FOR CONVENTIONAL HELICAL GEARS - ISO 53 CUTTER (m_G = 4.0, x_p = x_G = 0.0)

4.2 Development of the I Factor Charts

As a result of the rationalisation of the variables, then for a given gear ratio, m_G, equation (4.1) becomes,

$$f(N_P, \Psi_s, I) = 0 \qquad (4.2)$$

A normal intercept chart may be constructed from equation (4.2). Figure 4.1 shows one such chart from those produced for a series of gear ratios.

An examination of the charts developed showed that for a change in gear ratio, there was a minimal change in the curvature of the line from chart to chart for a particular number of pinion teeth. It was realised that if the movement between corresponding lines of pinion teeth for each value of helix angle was approximately constant, then a multiplying chart could be produced, thus limiting the number of charts to two for each addendum modification.

In general, equation (4.2) may be represented by:

$$I = I(N_P, m_G = m'_G, \Psi_s)\ M(N_P, m_G, \Psi_s) \qquad (4.3)$$

where m'_G = a constant

A major statistical analysis has shown, however, that equation (4.3) may be approximated by,

$$I = I(N_P, m_G = m'_G, \Psi_s)\ M(N_P, m_G) \qquad (4.4)$$

$$\text{where } M(N_P, m_G) = \left[\sum_{i=1}^{n+1} \frac{I(N_P, m_G, \Psi_{si})}{I(N_P, m_G = m'_G, \Psi_{si})} \right] / (n+1)$$

$$\text{and} \quad \Psi_{si} = \Psi_{s(min)} + [\Psi_{s(max)} - \Psi_{s(min)}].(i - 1)/n$$

Each of the variables $\Psi_{s(min)}$, $\Psi_{s(max)}$ and n are defined on the following page.

$\Psi_{s(min)}$ = minimum desired helix angle on chart to be generated

$\Psi_{s(max)}$ = maximum desired helix angle on chart to be generated

and n = number of increments between $\Psi_{s(min)}$ and $\Psi_{s(max)}$

The mathematical form previously described, lends itself to the development of a dual chart system for the graphical representation of the geometry factor for pitting resistance for conventional helical gears. The first chart represents $I(N_p, m_G = m'_G, \Psi_s)$ and the second chart represents $M(N_P, m_G)$. These charts have been called the reference and multiplier charts respectively.

The reference charts were developed in the form of an intercept chart using computer graphics for a constant gear ratio of $m_G = 2.5$. Examples of these reference charts are given in Figures 4.2 and 4.4.

The multiplier charts were developed using a computer program which generated five arrays. The flow for the computer program is given in ASME Paper 84-DET-167(14). These five arrays were the number of pinion teeth varying from 10 to 1,000, the gear ratio varying from 1 to 10, the geometry factor for pitting resistance at a constant gear ratio of $m_G = 2.5$, the geometry factor for pitting resistance at other values of gear ratio and the corresponding value of the multiplication factor for a given gear ratio and number of pinion teeth with the helix angle averaged out. The multiplier charts associated with Figures 4.2 and 4.4 are given as Figures 4.3 and 4.5 respectively.

To illustrate how the above algorithm was utilised to produce the pitting geometry factor multiplier, M_I, the following analysis indicates the calculations involved in finding the numerical value of one co-ordinate of Figure 4.3.

Utilising an ISO 53 (6) cutter with no addendum modifications, the pitting resistance geometry factor, I, can be calculated from the equations of Section 6.2 of AGMA 218.01 for any combination of pinion teeth, N_P, gear ratio, m_G, and helix angle, ψ.

Selecting a reference gear ratio, m'_G of 2.5, a constant number of pinion teeth of 40 and varying the helix angle from 5^o to 45^o, nine values of the reference pitting resistance geometry factor, [I], were calculated and are tabulated in Table 4.1.

i	m'_G	N_P	ψ_{si}	[I]
1	2.5	40	5	0.1946
2	2.5	40	10	0.1948
3	2.5	40	15	0.1953
4	2.5	40	20	0.1959
5	2.5	40	25	0.1965
6	2.5	40	30	0.1970
7	2.5	40	35	0.1972
8	2.5	40	40	0.1969
9	2.5	40	45	0.1960

TABLE 4.1 - REFERENCE PITTING RESISTANCE GEOMETRY FACTOR [I]

Consider the same set of calculations but changing the gear ratio, m_G to 5. The calculated values of I are tabulated in Table 4.2.

i	m_G	N_P	ψ_{si}	I
1	5.0	40	5	0.2307
2	5.0	40	10	0.2309
3	5.0	40	15	0.2313
4	5.0	40	20	0.2318
5	5.0	40	25	0.2322
6	5.0	40	30	0.2324
7	5.0	40	35	0.2323
8	5.0	40	40	0.2314
9	5.0	40	45	0.2301

TABLE 4.2 - PITTING RESISTANCE GEOMETRY FACTOR I

The multiplying factor, $M(N_P, m_G) = M(40,5)$, can now be calculated from the ratio I:[I]. The values obtained are tabulated in Table 4.3.

i	I	[I]	$M_I = I/[I]$
1	0.2307	0.1946	1.1855
2	0.2309	0.1948	1.1853
3	0.2313	0.1953	1.1843
4	0.2318	0.1959	1.1833
5	0.2322	0.1965	1.1817
6	0.2324	0.1970	1.1797
7	0.2323	0.1972	1.1780
8	0.2314	0.1969	1.1752
9	0.2301	0.1960	1.1740
			Σ 10.6270

TABLE 4.3 - PITTING RESISTANCE GEOMETRY MULTIPLIER M_I

Hence with $\Psi_{si} = 5 + [45-5].(i-1)/8$

$$M(40,5) = \left[\sum_{i=1}^{i=9} \frac{I(40,\ 5,\ \Psi_{si})}{I(40,\ 2.5,\ \Psi_{si})} \right] / \ 9$$

$$= \frac{10.6270}{9}$$

$$= 1.1808$$

This co-ordinate may be found on Figure 4.3.

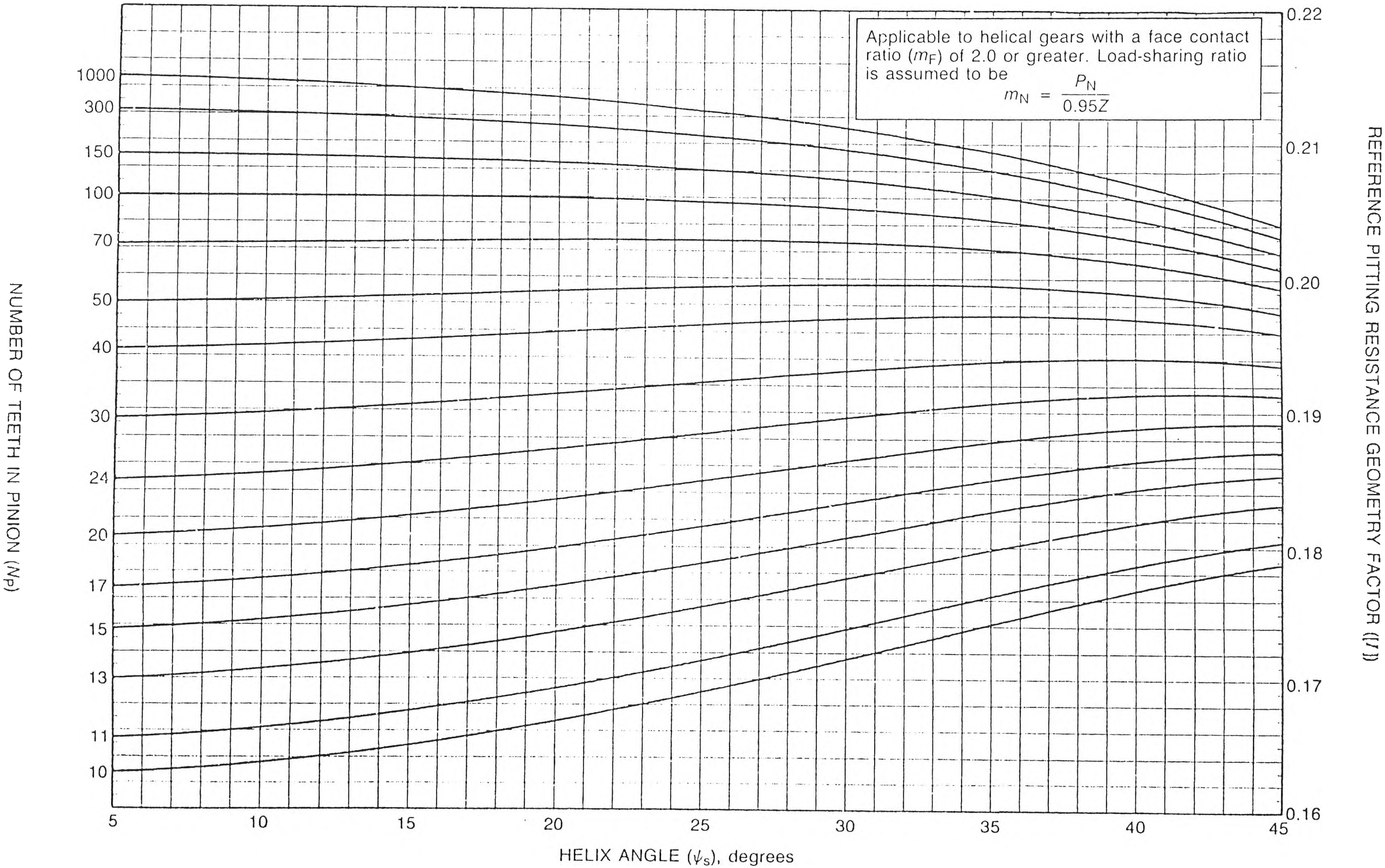


FIG. 4.2[I] FOR CONVENTIONAL HELICAL GEARS - ISO 53 CUTTER (x_p = 0.0, x_G = 0.0)

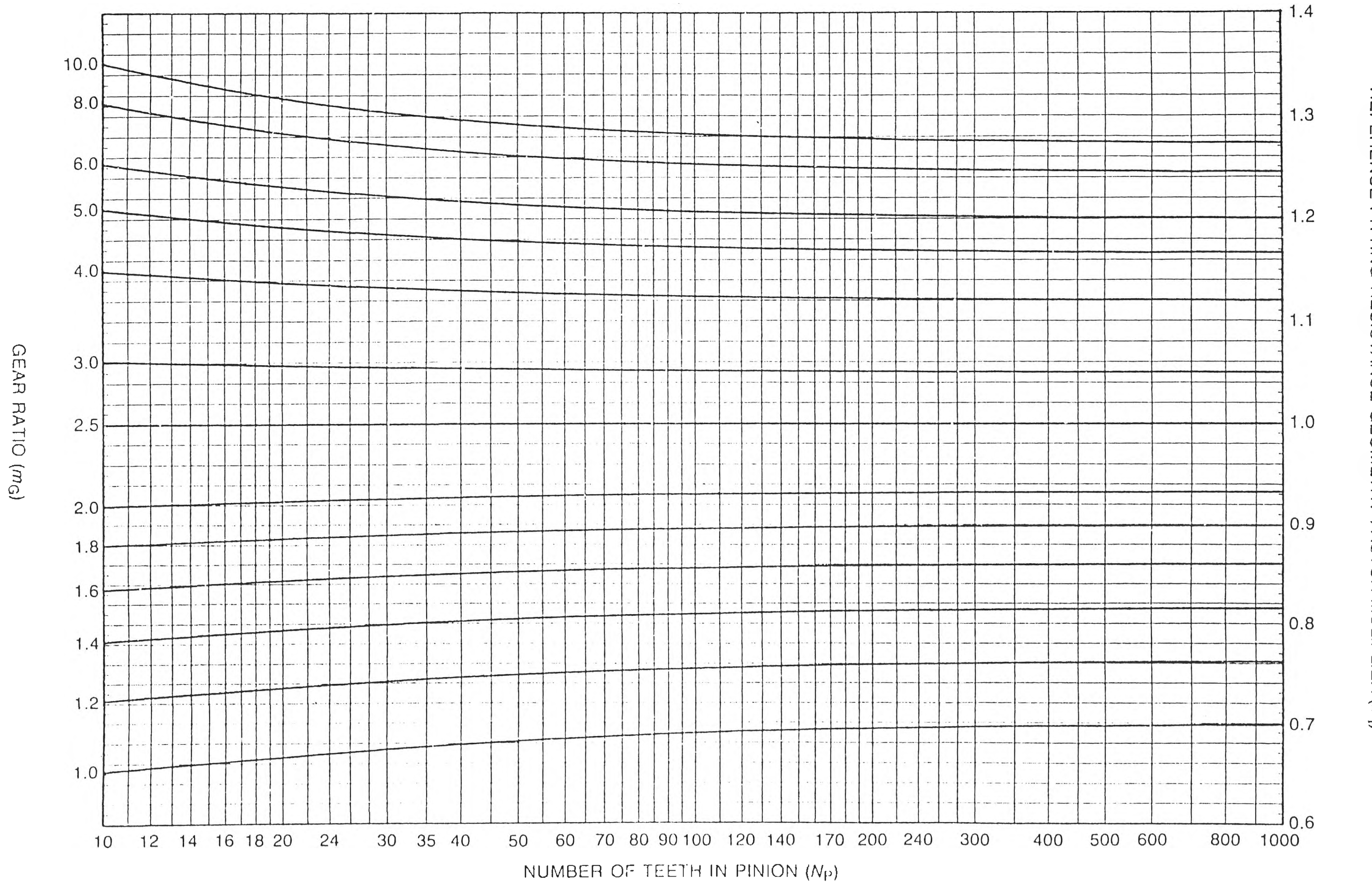


FIG. 4.3 M_I FOR CONVENTIONAL HELICAL GEARS - ISO 53 CUTTER (x_p = 0.0, x_G = 0.0)

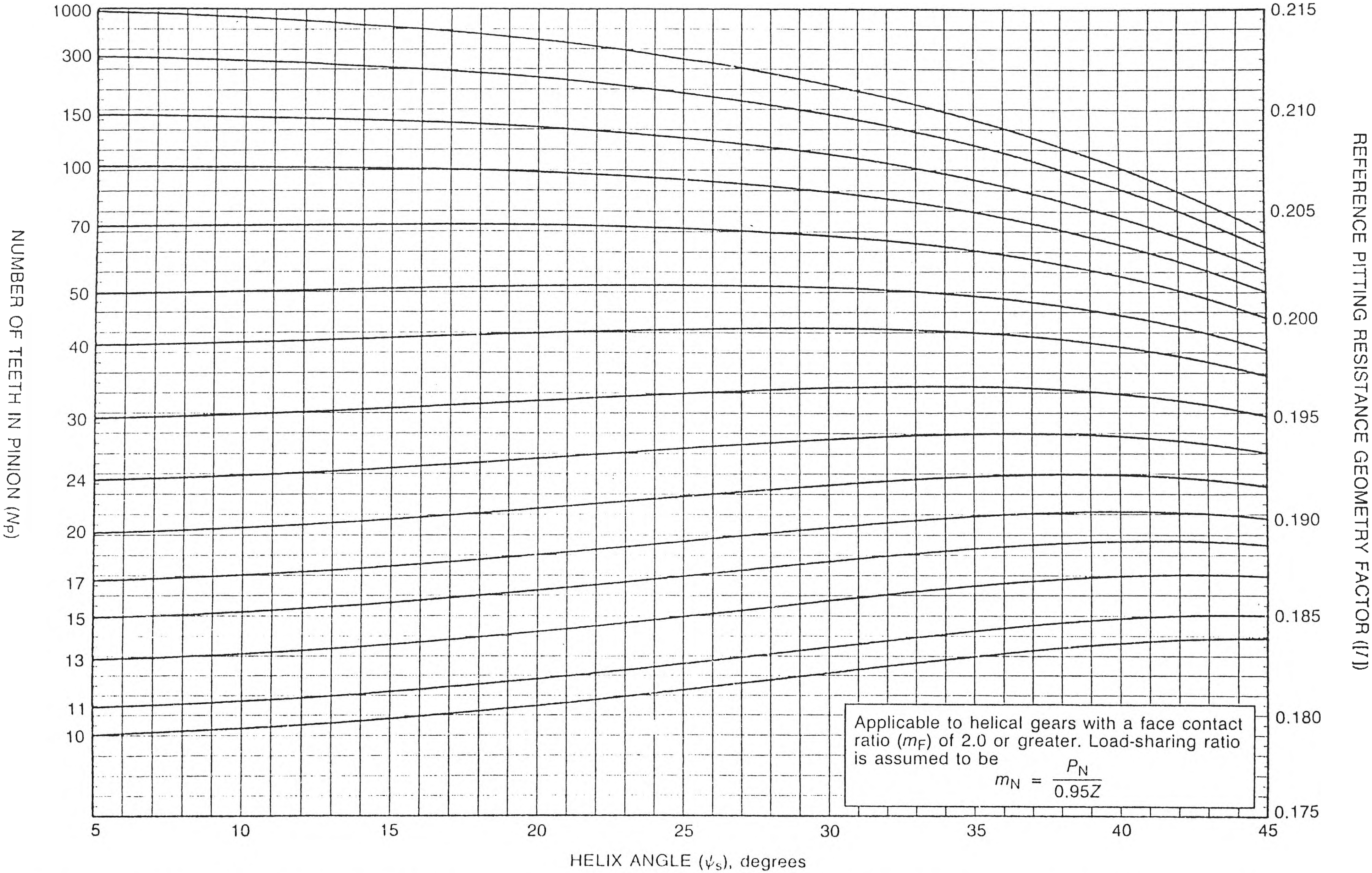


FIG. 4·4 [I] FOR CONVENTIONAL HELICAL GEARS - ISO 53 CUTTER (x_p = 0·2, x_G = 0·2)

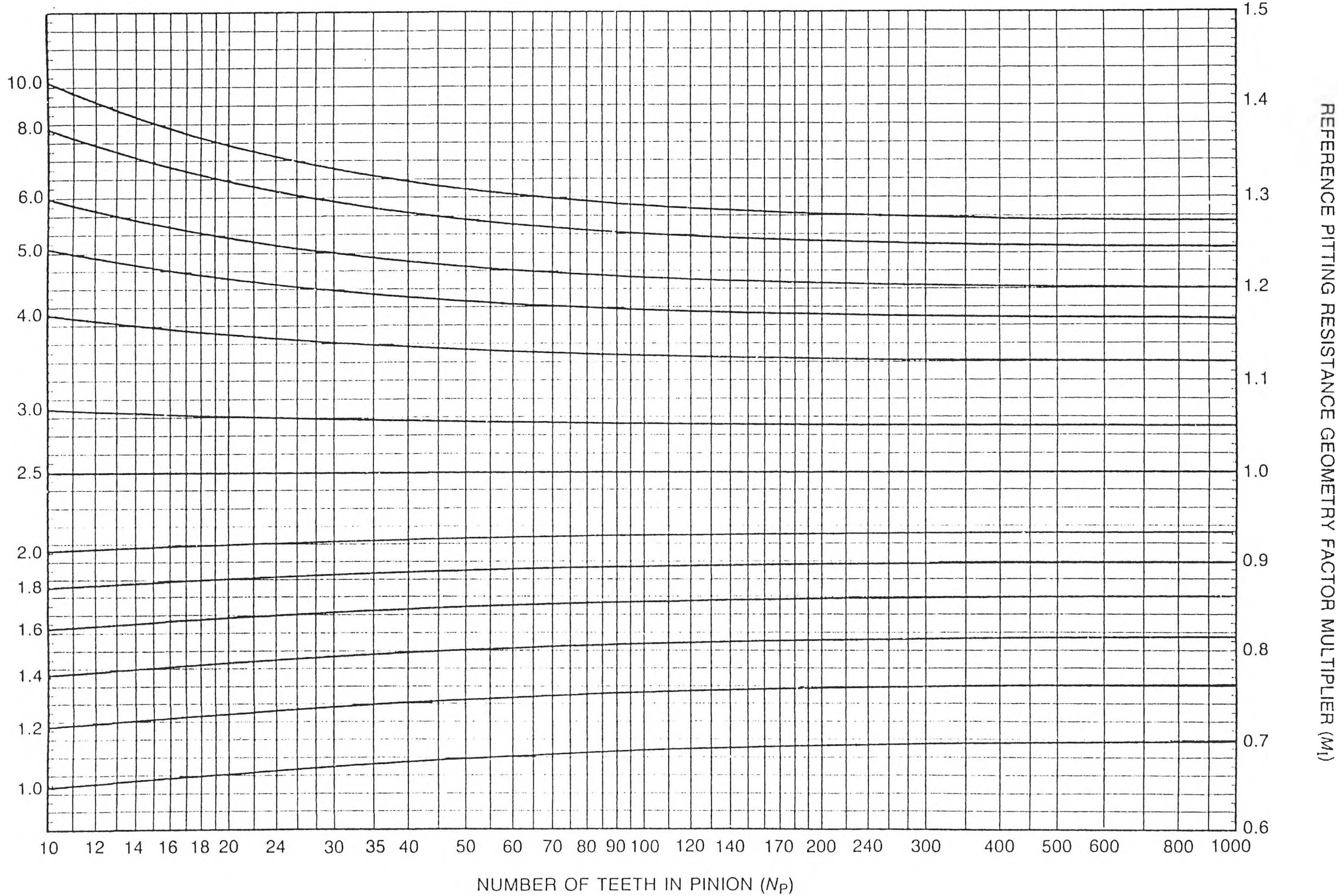


FIG. 4.5 M_I FOR CONVENTIONAL HELICAL GEARS - ISO 53 CUTTER ($x_p = 0{\cdot}2$, $x_G = 0{\cdot}2$)

4.3 Examples of the Use of the Design Charts

Two examples are presented to demonstrate the versatility of the charts.

Further, the results obtained for the geometry factor for pitting resistance using the design charts are compared with those calculated using the exact mathematical method of AGMA 218.01. This comparison demonstrates the high order of accuracy which can be achieved using the design charts.

The first example is for a pair of conventional helical gears running at standard centres.

Description	Pinion	Wheel
Number of teeth	40	200
Addendum modification coefficient	0.0	0.0
Helix angle, degrees	30.0	30.0
Cutter	ISO 53	ISO 53
Normal module, mm	8.0	8.0
Facewidth, mm	200.0	200.0

The following three steps calculate I using the charts:

$[I] = 0.197$ (Figure 4.2)

$M_I = 1.180$ (Figure 4.3)

$I = 0.197 \times 1.180 = 0.232$

The value of I calculated using the approximation for the load sharing ratio is I = 0.232, whilst the exact mathematical solution yields I = 0.244.

Thus the value of I obtained using the charts represents an error of 4.9%, due mainly to the approximation for m_N.

The second example is for a set of conventional helical gears running at extended centres.

Description	Pinion	Wheel
Number of teeth	40	200
Addendum modification coefficient	0.2	0.2
Helix angle, degrees	30.0	30.0
Cutter	ISO 53	ISO 53
Normal module, mm	8.0	8.0
Facewidth, mm	200.0	200.0

The following steps calculate I using the charts:

$[I] = 0.199$ (Figure 4.4)

$M_I = 1.192$ (Figure 4.5)

$I = 0.199 \times 1.192 = 0.237$

The value of I calculated using the approximation for the load sharing ratio is I = 0.237, whilst the exact mathematical solution yields I = 0.249.

Thus the value of I obtained using the charts represents an error of 4.8%, due mainly to the approximation for m_N.

4.4 Statistical Analysis of the Design Charts

Equation (4.4) is an approximation and thus a difference exists between the value obtained for the geometry factor for pitting resistance using the design charts and the exact mathematical solution. A comparison between values obtained using equation (4.4) and the mathematical method (incorporating equation (6.27) of AGMA 218.01) was carried out to give percentage difference as:

$$\% \text{ difference} = [(I_{Eq.(4.4)} - I_{actual})/I_{actual}] \times 100 \qquad (4.5)$$

This comparison was carried out with the aid of a SAS Institute statistical program which compared 96,750 combinations of gear ratio, number of pinion teeth and helix angle for varying addendum modifications. The results obtained are summarised in Table 4.4.

Combinations tested	96,750
Maximum positive error	+ 8.2%
Maximum negative error	- 5.6%
Percentage of Combinations Tested	**Percentage Range of Error**
99.96%	± 5.0%
98.84%	± 2.0%
93.71%	± 1.0%

TABLE 4.4 - COMPARISON OF DESIGN CHARTS AND EXACT MATHEMATICAL SOLUTION FOR I

SECTION 5

MATHEMATICAL CONUNDRUMS OF AGMA 218.01

5.1 Introduction

As a result of analysing AGMA 218.01 (4) for compatibility with the new Australian standard for the rating of gears, it became necessary to produce a number of computer programs for the analysis of geometry factors. In the testing of these programs several conundrums were found, two of which are highlighted in this Section. However, on the positive side, computer programs have been developed to the stage where they produce "contour plots" of geometry factors, thus enabling a gear designer to maximise the power rating of a particular gear set.

In rating a pair of gears to AGMA 218.01, the pitting resistance power rating, P_{ac}, is proportional to Id^2 and the bending strength power rating, P_{at}, is proportional to Jd, all other variables being constant for a specified application and material selection. Hence the conundrum is set to the gear designer of how to maximise the geometry factors and hence maximise the power rating for a particular pair of gears.

This Section presents the philosophy involved in the correct choice of addendum modifications to maximise the geometry factors and consequently the power rating of a particular gear set. For the example cited, a 25% increase in power rating can be achieved by the correct choice of addendum modifications, at no additional manufacturing cost.

However, the gear designer should give due consideration to the slide to roll ratio, as explained in Section 7.

5.2 The Calculation of Geometry Factors

The equations for the calculation of the pitting resistance geometry factor, I, are detailed in Section 6.2 of AGMA 218.01. Due to the complexity of the variables involved a computer program is justified, and is readily written as all variables are defined by equations.

The procedure for the calculation of the bending strength geometry factor, J, is detailed in Section 6.3 of AGMA 218.01, requiring a generated layout of the tooth profile, which as such, does not readily lend itself to computerisation. However, Appendix E of AGMA 218.01, offers an apercu of an analytical method for the calculation of J as proposed by Errichello (3). This technique has been modified by Davey (8) to encompass a reduction in the iteration procedure. A computer program has been written utilising the Lambda Method to calculate J, a listing of which may be found in Appendix A.

From an examination of the large number of variables involved in the calculation of I and J, it is not immediately apparent as to how to maximise the calculated values. However, in the context of this Section, it is assumed that the gear designer has selected the number of pinion and wheel teeth, module, facewidth, material and the helix angle if applicable. Hence the only choice of variables available to the gear designer is that of addendum modification coefficients and the cutter.

5.3 Addendum Modification and Cutter Selection

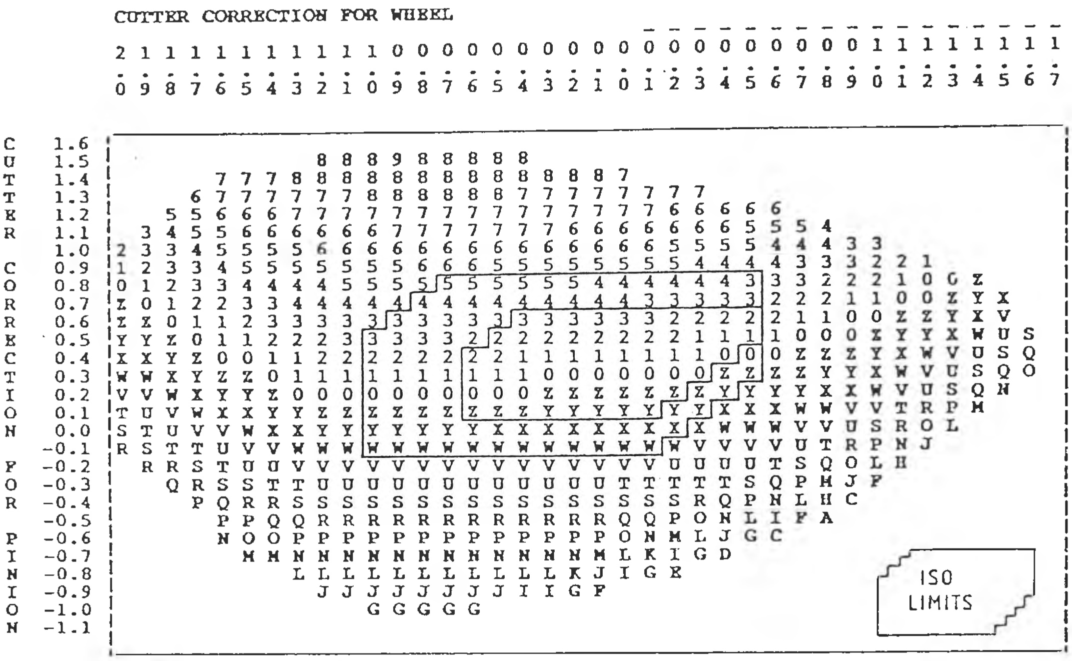
Guidance in this subject is offered in BSI PD 6457 (15), ISO/TR 4467(E) (7) and ANSI B6.1 (AGMA 201.02) (16). However, if safeguards are employed in the analysis, the selection of suitable coefficients can be extended beyond the limits suggested by the above documents.

The computer program that has been developed by the author incorporates thirty five safeguards to limit the choice of addendum modification coefficients. These include tip to root trochoid interference, insufficient tip width and insufficient bottom clearance. The program automatically reduces the outside diameter of the relevant gear of the pair to eliminate the problem. Further, limits are placed on the ratio of the approach path to the recess path, the slide to roll ratio and the total contact ratio. The limits that were incorporated by the author in the production of Figures 5.1 to 5.3 are as tabulated in Table 5.1. Many of these limits can be varied to suit the experience and requirements of the gear designer. It should be noted that the alphabetical symbols indicate that the particular addenda combination being tested is terminated, whilst the phonetical symbols indicate initial problems which have been rectified to enable completion of the analysis.

The choice of cutter is somewhat arbitrary, and is usually dictated by availability. However, it is recommended that a standard cutter be employed, and the physical dimensions of the cutter be ascertained prior to analysis of the geometry factors. For the purpose of demonstrating the techniques involved in the calculation of geometry factors, an ISO 53 (6) cutter has been selected, being the cutter recommended by the new Australian Standard for Gear Design AS 2938(5).

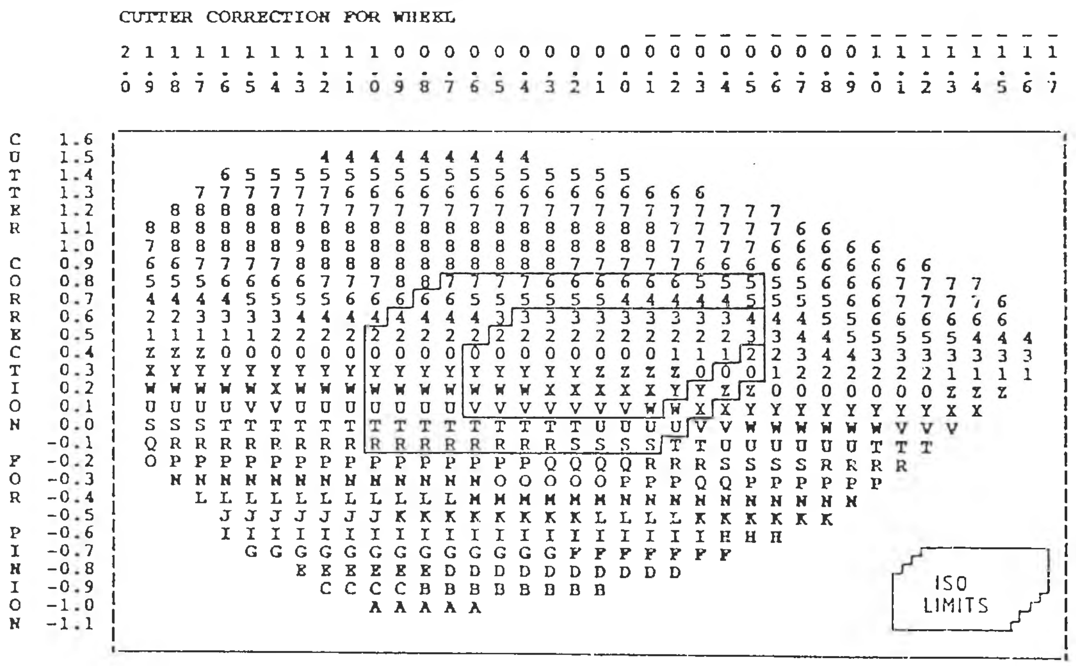
SYMBOL	SAFEGUARD	FIXED	VARIABLE	VALUE
A	Limits the maximum value of ϕ_t		YES	100 rad
B	Limits the minimum value of ϕ_t		YES	0.0 rad
C	Checks if ACOS (Tip angle ψ_{OP}) > 1.0	YES		
D	Checks if ACOS (Tip angle ψ_{OG}) > 1.0	YES		
E	Checks if $\Delta r_O > 1.0\ m_n$		YES	1.0
F	Checks if $\Delta R_o > 1.0\ m_n$		YES	1.0
G	Checks if $r < r_b$	YES		
H	Checks if $R < R_b$	YES		
I	Checks if $r_o < r_b$	YES		
J	Checks if $R_o < R_b$	YES		
K	Checks if $Z_b < 0.0$		YES	0.0 mm
L	Checks if $Z_a < 0.0$		YES	0.0 mm
M	Checks if $m_T < 1.0$		YES	1.0
N	Checks if $m_P < 1.1$		YES	1.1
O	Checks if $Z_c = f(\sqrt{}\ (\text{- ve}))$	YES		
P	Checks if $Z_{ch} = f(\sqrt{}\ (\text{- ve}))$	YES		
Q	Checks if $L_{min} < 0.0$		YES	0.0 mm
R	Checks if ACOS (Tip angle β_P) > 1.0	YES		
S	Checks if ACOS (Tip angle β_G) > 1.0	YES		
T	Checks if λ_P iteration exceeds 20		YES	20 loops
U	Checks if λ_G iteration exceeds 20		YES	20 loops
V	Checks if K_{6P} iteration exceeds 20		YES	20 loops
W	Checks if K_{6G} iteration exceeds 20		YES	20 loops
X	Checks if $C_{hP} = f\ (\sqrt{}\ (\text{- ve}))$	YES		
Y	Checks if $C_{hP} = f\ (\sqrt{}\ (\text{- ve}))$	YES		
#	Notification of undercut pinion	YES		
=	Notification of undercut wheel	YES		
@	Pinion bottom clearance initially < 0.1 m_n		YES	0.1
$	Wheel bottom clearance initially < 0.1 m_n		YES	0.1
(	Pinion tip width initially < 0.2 m_n		YES	0.2
)	Wheel tip width initially < 0.2 m_n		YES	0.2
>	Notification of $\lvert \epsilon_P - \epsilon_G \rvert > 0.2$		YES	0.2
?	Notification of $Z_a < Z_b$	YES		
*	Notification of initial pinion tip interference	YES		
+	Notification of initial wheel tip interference	YES		

TABLE 5.1 - COMPUTER SAFEGUARDS TO LIMIT ADDENDA MODIFICATIONS



A = 0.0226	G = 0.0420	M = 0.0615	S = 0.0809	Y = 0.1003	4 = 0.1197
B = 0.0259	H = 0.0453	N = 0.0647	T = 0.0841	Z = 0.1035	5 = 0.1229
C = 0.0291	I = 0.0485	O = 0.0679	U = 0.0873	0 = 0.1068	6 = 0.1262
D = 0.0323	J = 0.0517	P = 0.0712	V = 0.0906	1 = 0.1100	7 = 0.1294
E = 0.0356	K = 0.0550	Q = 0.0744	W = 0.0938	2 = 0.1132	8 = 0.1326
F = 0.0388	L = 0.0582	R = 0.0776	X = 0.0970	3 = 0.1165	9 = 0.1359

FIG. 5·1 CONTOUR PLOT OF I



A = 0.1309	G = 0.2082	M = 0.2855	S = 0.3628	Y = 0.4401	4 = 0.5174
B = 0.1438	H = 0.2211	N = 0.2984	T = 0.3757	Z = 0.4530	5 = 0.5303
C = 0.1567	I = 0.2340	O = 0.3113	U = 0.3886	0 = 0.4659	6 = 0.5432
D = 0.1696	J = 0.2469	P = 0.3242	V = 0.4015	1 = 0.4788	7 = 0.5561
E = 0.1825	K = 0.2598	Q = 0.3371	W = 0.4144	2 = 0.4917	8 = 0.5690
F = 0.1953	L = 0.2726	R = 0.3499	X = 0.4272	3 = 0.5045	9 = 0.5818

FIG. 5·2 CONTOUR PLOT OF J_p

5.4 Examples of the Use of the Computer Program

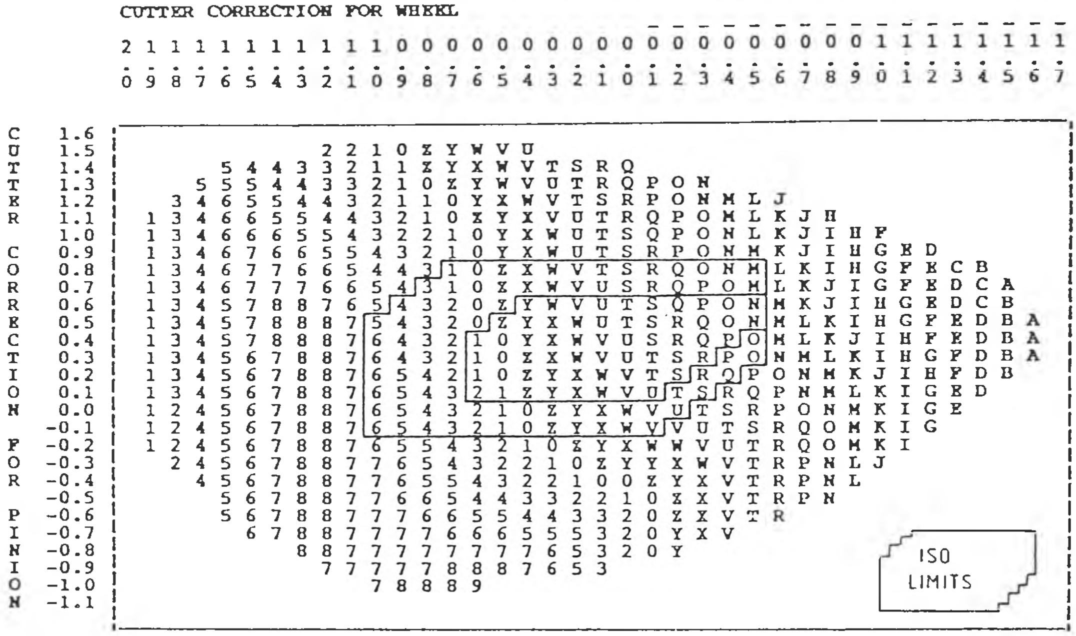
For the purpose of demonstration, assume a spur gear design has been formalised which has 25 and 50 pinion and wheel teeth respectively and is manufactured utilising a 2 module ISO 53 cutter and has no addendum modifications. Depending on the choice of materials and application factors, this design will have a particular power rating. Maintaining the same materials and application factors, this power rating can be increased if I and J can be increased. For optimum design, the ratio of the increases in I, J_P and J_G should be related to the original calculated powers; ie, if in the original design the strength power rating of the pinion was greater than that of the wheel, then $J_G:J_P$ should be increased in the same ratio as $P_{atP}:P_{atG}$.

The program provides "contour plots" of I, J_P and J_G, over a range of addendum modification coefficients. Further, all reasons for rejection are tabulated, together with notification of potential problem areas that have been left to the discretion of the gear designer for acceptance or rejection.

Having plotted all acceptable permutations within the nominated range of addendum modification coefficients, the program then selects five combinations which yield respectively the maximum I, J_P, J_G, $J_P + J_G$ and $I + J_P + J_G$. However, depending on the criteria for optimisation, the gear designer may select other combinations from the "contour plots".

To quantify the analysis, assume the original design was optimised to have equal power ratings for both wear and strength. Hence the gear designer requires the addendum modifications which will yield the maximum increase in $I + J_P + J_G$.

The "contour plots" for I, J_P and J_G, based on the original design are shown in Figures 5.1, 5.2 and 5.3 respectively, whilst the reasons for rejection and notification of potential problem areas are shown in Figure 5.4.



A = 0.1737	G = 0.2435	M = 0.3134	S = 0.3832	Y = 0.4531	4 = 0.5230
B = 0.1853	H = 0.2552	N = 0.3250	T = 0.3949	Z = 0.4647	5 = 0.5346
C = 0.1970	I = 0.2668	O = 0.3367	U = 0.4065	0 = 0.4764	6 = 0.5462
D = 0.2086	J = 0.2785	P = 0.3483	V = 0.4182	1 = 0.4880	7 = 0.5579
E = 0.2203	K = 0.2901	Q = 0.3600	W = 0.4298	2 = 0.4997	8 = 0.5695
F = 0.2319	L = 0.3018	R = 0.3716	X = 0.4415	3 = 0.5113	9 = 0.5812

FIG. 5.3 CONTOUR PLOT OF J_G

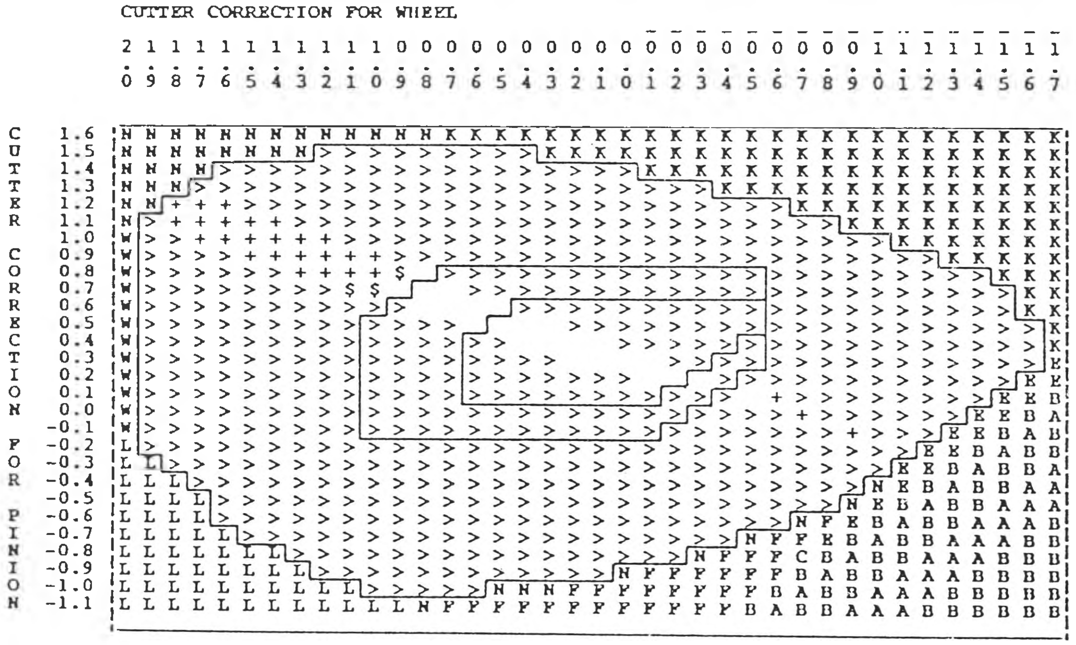


FIG. 5.4 CONTOUR PLOT OF ACCEPTABLE SOLUTIONS

For a design that maintains the existing gear ratio, but extends the centre distance, the contours show that significant increases can be achieved, a fact that has long been known but has previously been difficult to quantify. The program suggests addendum modifications coefficients of $x_P = 1.0$ and $x_G = 1.4$ to maximise $I + J_P + J_G$, with approximately 0.4 mm of topping on both the pinion and wheel. However, to integerise the centre distance x_P and x_G have been selected at 0.8441 and 1.5 respectively. From Table 5.2, this design shows a 32.5% increase in power rating over the original design with a 4mm increase in centre distance.

In many instances, the gear centres cannot be extended. However, a small variation in gear ratio can usually be tolerated. Hence consider the choice of 24 pinion teeth and 47 wheel teeth, resulting in a 2% variation in gear ratio. In this instance, the program suggests $x_P = 0.9608$ and $x_G = 1.4$ to maintain the 75mm centre distance, with approximately 0.45mm of topping on both the pinion and wheel. From Table 5.2, this design shows a 25.4% increase in power rating over the original design at no additional cost to the manufacturer.

Appreciating that a solution which maintains the existing gear ratio and centre distance is not always possible, the selection of 24 pinion teeth and 48 wheel teeth will suffice in this example. Addendum modifications of $x_P = 0.678$ and $x_G = 1.030$ are suggested with approximately 0.15mm of topping on both the pinion and wheel. From Table 5.2, this design shows a 21.2% increase in power rating over the original design at no additional cost to the manufacturer.

A summary of the computer output is contained in Table 5.2. The major variables associated with the decision as to whether to accept or reject a particular set of addendum modification coefficients have been included. The power increase for pitting resistance has been referenced back to the original gear design, based on equation (5.1).

DESCRIPTION		REFERENCE	DESIGN 1	DESIGN 2	DESIGN 3
Number of Pinion Teeth		25	25	24	24
Number of Wheel Teeth		50	50	47	48
Pinion Addendum Modification Coefficient		0.000	0.844	0.961	0.678
Wheel Addendum Modification Coefficient		0.000	1.500	1.400	1.030
Pinion Topping	(mm)	0.000	0.425	0.445	0.150
Wheel Topping	(mm)	0.000	0.425	0.445	0.150
Pinion Backlash	(mm)	0.048	0.048	0.048	0.048
Wheel Backlash	(mm)	0.048	0.048	0.048	0.048
Module	(mm)	2	2	2	2
Centre Distance	(mm)	75	79	75	75
Gear Ratio		2	2	1.96	2
Clearance of Pinion	(mm)	0.570	0.310	0.298	0.305
Clearance of Wheel	(mm)	0.570	0.310	0.298	0.305
Pinion Tip Width	(mm)	1.384	1.240	1.126	1.017
Wheel Tip Width	(mm)	1.498	1.140	1.195	1.129
Approach Length	(mm)	5.180	3.964	3.520	4.116
Recess Length	(mm)	4.758	3.813	4.111	4.428
Pinion Slide/Roll Ratio		1.156	0.572	0.655	0.775
Wheel Slide/Roll Ratio		2.305	0.749	0.660	0.925
Contact Ratio		1.683	1.317	1.293	1.447
Pitting Resistance Geometry Factor, I		0.099	0.120	0.123	0.120
Bending Strength Geometry Factor, J_P		0.395	0.539	0.557	0.525
Bending Strength Geometry Factor, J_G		0.435	0.547	0.538	0.528
Pinion Pitch Diameter, d	(mm)	50.000	52.667	50.704	50.000
Pitting Resistance Power Increase %		0.0	34.5	27.8	21.2
Strength Power Increase %		0.0	32.5	25.4	21.4

TABLE 5.2 - SUMMARY OF COMPUTER ANALYSIS

$$\text{Increase in Pitting Resistance} = \left[\frac{I}{I_s} \left[\frac{d}{d_s} \right]^2 - 1.0 \right] \times 100 \qquad (5.1)$$

Similarly, the power increase for strength is based on equation (5.2).

$$\text{Increase in Strength} = \left[\frac{Jd}{J_s\, d_s} - 1.0 \right] \times 100 \qquad (5.2)$$

where I = geometry factor for pitting resistance

J = geometry factor for bending strength

d = operating pitch diameter of pinion, mm

Subcript s = reference

5.5 LCR and Conventional Helical Gears

The techniques which have been developed in the previous section can also be applied to low contact ratio (LCR) and conventional helical gears. AGMA 218.01 defines LCR gears as those having a face contact ratio less than or equal to unity. For the gear set detailed in Tables 5.3 and 5.4, the facewidth which yields the transition from a LCR to a conventional helical gear is 87.804mm.

An examination of Table 5.3 shows that for a facewidth of 87mm, the gear set is analysed as a LCR and yields $J_P = 0.637$. Similarly from Table 5.4, with a facewidth of 88mm, the gear set is analysed as a conventional helical gear and yields $J_P = 0.609$. In this example, the bending strength power rating is proportional to FJ. Hence one may hypothesise that the strength rating of the pinion has been increased by 3.6% as a result of a reduction in facewidth.

This example serves to demonstrate that, as with any computerised analysis, the gear designer must be careful as to the interpretation of the results. It should be noted that I and J_G exhibit a smooth transition as the width of the gear forces the analysis through that of a LCR to a conventional helical gear.

STEP	NAME	PINION	WHEEL	UNITS	DESCRIPTION
I	NT	23	84		NUMBER OF TEETH
N	PHIC	20 00'00"	20 00'00"	DEG	NORMAL PRESSURE ANGLE
P	MN	8.00000	8.00000	MM	NORMAL METRIC MODULE
U	PSIS	16 37'58"	16 37'58"	DEG	HELIX ANGLE AT REF PITCH DIA
T	X	0.29000	0.13420		ADDENDUM MODIFICATION COEFF
	F	87.00000	87.00000	MM	NET FACE WIDTH
D	RT	3.04000	3.04000	MM	TIP RADIUS OF CUTTING TOOL
A	HA	10.00000	10.00000	MM	STANDARD ADDENDUM OF TOOL
T	HB	8.00000	8.00000	MM	STANDARD DEDENDUM OF TOOL
A	DLTARO	0.00000	0.00000	MM	TRUNCATION APPLIED
	BN	0.18150	0.27910	MM	BACKLASH APPLIED
	QV	6	6		AGMA 390 QUALITY NUMBER
1	PHIS	20 48'00"	20 48'00"	DEG	TRANSVERSE PRESSURE ANGLE
2	PHIT	21 52'58"	21 52'58"	DEG	OPERATING TRANSVERSE PRESS ANG
3	C	450.00054	450.00054	MM	OPERATING CENTRE DISTANCE
4	RB	89.75960	327.81768	MM	BASE RADII
5	R	96.72909	353.27145	MM	OPERATING PITCH RADII
6	RO	106.33747	359.74609	MM	TIP RADII
7	MF	0.99084	0.99084		FACE CONTACT RATIO
8	ZB	16.49749	16.49749	MM	LENGTH OF APPROACH PATH
8	ZA	20.96464	20.96464	MM	LENGTH OF RECESS PATH
9	Z	37.46213	37.46213	MM	LENGTH OF LINE OF ACTION
10	ZC	3.55607	3.55607	MM	DIST STRESS TO PITCH POINTS
11	CX	0.92571	0.92571		CONTACT HEIGHT FACTOR
12	ZCH	-4.01454	-4.01454	MM	LENGTH OF LINE OF ACTION LCR
12	CXH	1.07747	1.07747		CONTACT HEIGHT FACTOR LCR
13	PSIB	15 36'12"	15 36'12"	DEG	BASE HELIX ANGLE
14	CPSI	1.35593	1.35593		HELICAL OVERLAP FACTOR
15	CC	0.13576	0.13576		CURVATURE FACTOR
16	LMIN	87.00000	87.00000	MM	MIN LENGTH OF LINES OF CONTACT
17	MNN	1.00000	1.00000		LOAD SHARING RATIO
18	I	0.23105	0.23105		GEOMETRY FACTOR - PITTING
19	PSI	16 44'57"	16 44'57"	DEG	OPERATING PITCH DIA HELIX ANG
20	PHIN	21 02'14"	21 02'14"	DEG	OPERATING NORMAL PRESSURE ANG
21	NE	26.19441	95.66655		VIRTUAL NUMBER OF SPUR TEETH
21	ROE	115.09765	391.73981	MM	VIRTUAL SPUR TIP RADII
22	BETA	0.54440	0.40797	RAD	BASE & TIP RADII INCLUDED ANGS
22	PHILN	0.52337	0.40024	RAD	LOAD ANGLE AT TIP
22	RBE	98.45879	359.58863	MM	EQUIVALENT BASE RADII
22	RE	105.48993	385.26756	MM	EQUIVALENT OPERATING RADII
22	ZNE	38.86333	38.86333	MM	EQUIVALENT Z FOR LCR HELICAL
22	PHIL	0.36852	0.35784	RAD	LOAD ANGLES AT HPSTC
23	LAMBAI	0.34907	0.34907	RAD	LAMBDA - INITIAL VALUE
24	K1	0.61117	0.78373	MM	CONSTANT - K1
24	K2	1.50644	1.50644	MM	CONSTANT - K2
24	K3	21.42984	61.03314		CONSTANT - K3
25	K4	0.92064	1.06095		VARIABLE - K4
25	K5	0.87959	1.04385	RAD	VARIABLE - K5
25	K6	0.86582	1.03346	RAD	VARIABLE - K6
25	K7	0.86580	1.03345	RAD	VARIABLE - K7
25	K8	0.16986	0.05899	RAD	VARIABLE - K8
25	K9	1.32311	1.91114	MM	VARIABLE - K9
26	TE	17.21445	17.97660	MM	INSCRIBED PARABOLA - WIDTHS
26	HE	7.67291	8.94475	MM	INSCRIBED PARABOLA - HEIGHTS
27	LAMBDA	0.51116	0.46559	RAD	LAMBDA - FINAL VALUES
28	CH	1.00000	1.00000		HELICAL FACTOR
29	KPSI	1.00000	1.00000		HELIX ANGLE FACTOR
30	Y	0.94088	0.85741		TOOTH FORM FACTORS
31	B	8.64095	11.90878	MM	OPERATING DEDENDA
31	RF	3.32421	3.24089	MM	MINIMUM ROOT FILLET RADII
31	H	0.17170	0.17170		DOLAN - BROGHAMER FACTOR - H
31	L	0.14170	0.14170		DOLAN - BROGHAMER FACTOR - L
31	M	0.46037	0.46037		DOLAN - BROGHAMER FACTOR - M
31	KF	2.00302	1.92958		STRESS CONCENTRATION FACTORS
32	J	0.63692	0.60250		GEOMETRY FACTORS - STRENGTH

TABLE 5.3 - GEOMETRY FACTORS FOR LCR GEARS

STEP	NAME	PINION	WHEEL	UNITS	DESCRIPTION
I	NT	23	84		NUMBER OF TEETH
N	PHIC	20 00'00"	20 00'00"	DEG	NORMAL PRESSURE ANGLE
P	MN	8.00000	8.00000	MM	NORMAL METRIC MODULE
U	PSIS	16 37'58"	16 37'58"	DEG	HELIX ANGLE AT REF PITCH DIA
T	X	0.29000	0.13420		ADDENDUM MODIFICATION COEFF
	F	88.00000	88.00000	MM	NET FACE WIDTH
D	RT	3.04000	3.04000	MM	TIP RADIUS OF CUTTING TOOL
A	HA	10.00000	10.00000	MM	STANDARD ADDENDUM OF TOOL
T	HB	8.00000	8.00000	MM	STANDARD DEDENDUM OF TOOL
A	DLTARO	0.00000	0.00000	MM	TRUNCATION APPLIED
	BN	0.18150	0.27910	MM	BACKLASH APPLIED
	QV	6	6		AGMA 390 QUALITY NUMBER
1	PHIS	20 48'00"	20 48'00"	DEG	TRANSVERSE PRESSURE ANGLE
2	PHIT	21 52'58"	21 52'58"	DEG	OPERATING TRANSVERSE PRESS ANG
3	C	450.00054	450.00054	MM	OPERATING CENTRE DISTANCE
4	RB	89.75960	327.81768	MM	BASE RADII
5	R	96.72909	353.27145	MM	OPERATING PITCH RADII
6	RO	106.33747	359.74609	MM	TIP RADII
7	MF	1.00223	1.00223		FACE CONTACT RATIO
8	ZB	16.49749	16.49749	MM	LENGTH OF APPROACH PATH
8	ZA	20.96464	20.96464	MM	LENGTH OF RECESS PATH
9	Z	37.46213	37.46213	MM	LENGTH OF LINE OF ACTION
10	ZC	-4.01454	-4.01454	MM	DIST STRESS TO PITCH POINTS
11	CX	1.07747	1.07747		CONTACT HEIGHT FACTOR
13	PSIB	15 36'12"	15 36'12"	DEG	BASE HELIX ANGLE
14	CPSI	1.00000	1.00000		HELICAL OVERLAP FACTOR
15	CC	0.13576	0.13576		CURVATURE FACTOR
16	LMIN	139.48110	139.48110	MM	MIN LENGTH OF LINES OF CONTACT
17	MNN	0.63091	0.63091		LOAD SHARING RATIO
18	I	0.23184	0.23184		GEOMETRY FACTOR - PITTING
19	PSI	16 44'57"	16 44'57"	DEG	OPERATING PITCH DIA HELIX ANG
20	PHIN	21 02'14"	21 02'14"	DEG	OPERATING NORMAL PRESSURE ANG
21	NE	26.19441	95.66655		VIRTUAL NUMBER OF SPUR TEETH
21	ROE	115.09765	391.73981	MM	VIRTUAL SPUR TIP RADII
22	BETA	0.54440	0.40797	RAD	BASE & TIP RADII INCLUDED ANGS
22	PHIL	0.52337	0.40024	RAD	LOAD ANGLES AT TIP
23	LAMBAI	0.26180	0.26180	RAD	LAMBDA - INITIAL VALUE
24	K1	0.61117	0.78373	MM	CONSTANT - K1
24	K2	1.50644	1.50644	MM	CONSTANT - K2
24	K3	21.42984	61.03314		CONSTANT - K3
25	K4	1.19398	1.27750		VARIABLE - K4
25	K5	1.14075	1.25691	RAD	VARIABLE - K5
25	K6	1.10251	1.23141	RAD	VARIABLE - K6
25	K7	1.10187	1.23113	RAD	VARIABLE - K7
25	K8	0.20713	0.07786	RAD	VARIABLE - K8
25	K9	1.73235	2.73235	MM	VARIABLE - K9
26	TE	16.32205	17.30213	MM	INSCRIBED PARABOLA - WIDTHS
26	HE	14.72394	14.59759	MM	INSCRIBED PARABOLA - HEIGHTS
27	LAMBDA	0.27035	0.28808	RAD	LAMBDA - FINAL VALUES
28	CH	1.31672	1.31672		HELICAL FACTOR
29	KPSI	0.91751	0.91751		HELIX ANGLE FACTOR
30	Y	0.57087	0.58778		TOOTH FORM FACTORS
31	B	8.64095	11.90878	MM	OPERATING DEDENDA
31	RF	3.32421	3.24089	MM	MINIMUM ROOT FILLET RADII
31	H	0.17170	0.17170		DOLAN - BROGHAMER FACTOR - H
31	L	0.14170	0.14170		DOLAN - BROGHAMER FACTOR - L
31	M	0.46037	0.46037		DOLAN - BROGHAMER FACTOR - M
31	KF	1.48551	1.54278		STRESS CONCENTRATION FACTORS
32	J	0.60910	0.60387		GEOMETRY FACTORS - STRENGTH

TABLE 5.4 - GEOMETRY FACTORS FOR CONVENTIONAL HELICAL GEARS

5.6 Stress Concentration Factor

Equation (6.60) of AGMA 218.01 defines the stress concentration factor, K_f, in terms of the operating dedendum and the operating normal pressure angle. If identical pinions are meshed with a series of wheels, each of which has a constant number of teeth, but a varying amount of addendum modification, then the operating dedendum of the pinions must change. Equation (6.43) of AGMA 218.01 then implies that the minimum fillet radius of the pinion, r_{fP}, will change, which is not the case. This conundrum is best explained if the definition of r_f were "the fillet radius at the point where the Lewis parabola is tangential to the root fillet".

Considering the analysis of a helical pinion, the only change in the point of the tangency of the Lewis parabola would be as a result of a change in the load angle, ϕ_{LN}, and/or a change in the equivalent number of pinion teeth, N_{eP}, as a result of a change in the operating helix angle, ψ. As the changes in ϕ_{LNP} and N_{eP} are both relatively small, one would expect minor changes in r_{fP}. However, as indicated in Table 5.5, this is not the case. This phenomenon would at first appear to be an error, but when considered in conjunction with the Dolan and Broghamer (12) factors, the subsequent numerical value of the strength geometry factor, J_P, is approximately constant.

N_P	N_G	x_P	x_G	Δr_o (mm)	ΔR_o (mm)	ϕ_n (deg)	ψ (deg)	ϕ_{LNP} (deg)	N_{eP}	t_{eP} (mm)	h_{eP} (mm)	C_h	Y_P	b_P (mm)	r_{fP} (mm)	K_{fP}	J_P
25	50	0.0	0.0	0.286	0.0	20.000	30.000	23.563	38.490	2.0296	1.4603	1.460	0.619	1.250	0.418	1.650	0.497
25	50	0.0	0.4	0.286	0.0	21.028	30.223	23.543	38.752	2.0311	1.4605	1.476	0.622	1.380	0.429	1.623	0.495
25	50	0.0	0.8	0.286	0.0	21.951	30.436	23.523	39.005	2.0326	1.4607	1.491	0.625	1.505	0.441	1.598	0.496
25	50	0.0	1.2	0.286	0.0	22.790	30.639	23.505	39.251	2.0340	1.4609	1.505	0.627	1.624	0.454	1.576	0.499

TABLE 5.5 - VARIATION OF ROOT FILLET RADIUS WITH OPERATING DEDENDUM

SECTION 6

GEAR POWER RATING TO AGMA 218.01

6.1 Introduction

This Section outlines the procedure used to rate gears to AGMA 218.01 (4). The power ratings obtained by the use of this standard form the basis of the gear rating package.

The power rating formulae described in AGMA 218.01 are applicable for rating the pitting resistance of internal and external spur and helical gear teeth and the bending strength of external spur and helical involute gear teeth. The formulae evaluate gear tooth capacity as influenced by the major factors which affect tooth pitting, and gear tooth fracture at the fillet radius. In the context of this Thesis, however, only external gears are considered.

Before proceeding, it is important to distinguish between the two major criteria that are used to determine the power transmission capability of a gear pair. The first criterion is a function of the Hertzian contact stress between the two contacting tooth surfaces. The ability of a gear to withstand this type of stress is termed its pitting resistance. The pitting of gear teeth is generally considered to be a fatigue phenomenon. Initial pitting and destructive pitting are illustrated and discussed in ANSI/AGMA 110(17). The pitting resistance power formula, equation (6.1), aims to determine a load rating at which destructive pitting does not occur during the design life. The ratings for pitting resistance are based on formulae developed by Hertz for contact pressure between two curved surfaces, modified for the effect of load sharing between adjacent teeth. The second criteria for determining the power rating of a gear pair is based upon a fatigue phenomenon related to the resistance to cracking at the tooth fillet. Typical cracks and fractures are illustrated in ANSI/AGMA 110. The basic theory employed in this analysis assumes the gear tooth to be rigidly fixed at its base and thus the critical stress occurs at the fillet. The intent of the AGMA strength rating formula is to determine the load which can be transmitted for the design life of the gear drive without causing root fillet cracking or failure.

6.2 Pitting Resistance Power Rating (P_{ac})

The pitting resistance power rating is obtained from AGMA 218.01, equation (5.5M) as

$$P_{ac} = \frac{n_p\ F}{1.91 \times 10^7} \quad \frac{I\ C_v}{C_s\ C_m\ C_f\ C_a} \left[\frac{d\ s_{ac}\ C_L\ C_H}{C_p\quad C_T\quad C_R} \right]^2 \qquad (6.1)$$

where P_{ac} = allowable transmitted power for pitting resistance, kW

n_p = pinion speed, RPM

F = net facewidth of narrowest member, mm

I = geometry factor for pitting resistance

C_v = dynamic factor for pitting resistance

d = operating pitch diameter of pinion, mm

s_{ac} = allowable contact stress number for gear material, MPa

C_L = life factor for pitting resistance

C_H = hardness ratio factor

C_s = size factor for pitting resistance

C_m = load distribution factor for pitting resistance

C_f = surface condition factor

C_a = application factor for pitting resistance

C_p = elastic coefficient, $(MPa)^{1/2}$

C_T = temperature factor for pitting resistance

C_R = reliability factor for pitting resistance

The ratings of both pinion and gear teeth must be calculated to evaluate differences in material properties and the number of tooth contact cycles under load. The pitting resistance power is based on the lower value of the product $s_{ac}.C_L.C_H$ for each of the mating gears.

6.3 Bending Strength Power Rating (P_{at})

The bending strength power rating is obtained from AGMA 218.01, equation (5.14M) as

$$P_{at} = \frac{n_p\ d\ K_v}{1.91 \times 10^7\ K_a}\ F\ m\ \frac{J}{K_s\ K_m}\ \frac{s_{at}\ K_L}{K_R\ K_T} \qquad (6.2)$$

where K_v = dynamic factor for bending strength

m = transverse module, mm

J = geometry factor for bending strength

s_{at} = allowable bending stress number for gear material, MPa

K_L = life factor for bending strength

K_a = application factor for bending strength

K_s = size factor for bending strength

K_m = load distribution factor for bending strength

K_R = reliability factor for bending strength

K_T = temperature factor for bending strength

The ratings of both pinion and gear teeth must be calculated to evaluate differences in geometry factors, number of load cycles, and material properties. The bending strength power rating is based upon the lower value of the product $s_{at}.K_L.J$ for each of the mating gears.

6.4 Geometry Factors (I) and (J)

The tenet of this Thesis is the presentation of an original analytical method for the calculation of the bending strength geometry factor (J) and a modified method for the calculation of the pitting resistance geometry factor (I), the procedural steps being documented in Section 1.9. However, for integrality, the basic equations are restated.

6.4.1 Pitting resistance geometry factor (I)

The pitting resistance geometry factor I, evaluates the radii of curvature of the contacting tooth profiles based on the pressure angle, helix angle, and gear ratio. Effects of addendum modification and load sharing are considered. The I factor is defined by AGMA 218.01, equation (6.2) as

$$I = \frac{C_c \ C_x \ C_\psi{}^2}{m_N} \tag{6.3}$$

where C_c = curvature factor at pitch line
C_x = contact height factor
C_ψ = helical overlap factor
m_N = load sharing ratio

6.4.2 Bending strength geometry factor (J)

The geometry factor J, considers the shape of the tooth, the position at which the critical load is applied, stress concentration due to geometric shape, and the sharing of load between oblique lines of contact in helical gears. Both the tangential (bending) and the radial (compressive) components of the tooth load are included.

The J factor is defined AGMA 218.01, equation (6.32) as

$$J = \frac{Y \ C_\psi}{K_f \ m_N} \tag{6.4}$$

where Y = tooth form factor
K_f = stress correction factor

6.5 Power Rating Modification Factors

The catalogue tabulations of power ratings for proprietary gearboxes are for combined design factors equalling 1 unless otherwise specified.

However, it is recognised that for the vast majority of industrial type gear systems, there is a need for the selection of application factors, life factors, reliability factors, etc. It will be seen from Clauses 8, 9, 11, 12, 13, 15, 16, 17 and 18 of AGMA 218.01, that the components are separated and identified to allow for each part to be carefully considered and assigned.

6.5.1 Dynamic factors (C_v) and (K_v)

The dynamic factors, C_v and K_v account for internally generated gear tooth loads which are induced by non-conjugate meshing action of the gear teeth. Even if the input torque and speed are constant, significant vibration of the gear masses, and therefore, dynamic tooth forces can exist. These forces result from relative displacements between the gears as they vibrate in response to an excitation known as "transmission error". The ideal kinematics of a gear pair require an exact ratio between the input and output rotation. Transmission error is defined as the departure from uniform relative angular motion of the meshing pair of gears.

The dynamic factors relate the total tooth load including dynamic effects to the transmitted tangential tooth load, and is defined by equation (8.1) of AGMA 218.01 as

$$C_v = K_v = W_t/(W_d + W_t) \tag{6.5}$$

where W_t = transmitted tangential load, N

W_d = incremental dynamic tooth load, N

The calculation of the incremental dynamic tooth load W_d, requires a detailed investigation of the system dynamics of the gear drive. Where it is not possible to determine the dynamic loadings, AGMA 218.01 suggests an alternate and approximate method, for the calculation of C_v and K_v. The method is based on the transmission accuracy level number, Q_v. Q_v can be the same as the quality number from AGMA 390.03(18) when manufacturing techniques ensure equivalent transmission accuracy, or when the pitch and profile accuracy are the same as AGMA 390 tolerances. The approximate method has been used in the gear rating package, and is described in Clause 8.3 of AGMA 218.01 as

For $Q_v < 6$:

$$C_v = K_v = 50/(50 + \sqrt{200}\ v_t) \qquad (6.6)$$

where v_t = pitch line velocity at operating pitch diameter, m/s.

$$v_t = \pi\ n_p\ d/60000 \qquad (6.7)$$

For $6 \leq Q_v \leq 11$:

$$C_v = K_v = \left[\frac{A}{A + \sqrt{200}\ v_t} \right]^B \qquad (6.8)$$

where $A = 50 + 56\ (1.0 - B)$ (6.9)

$$B = \frac{(12 - Q_v)^{0.667}}{4} \qquad (6.10)$$

For $Q_v \geq 12$, values of C_v and K_v between 0.90 and 0.98 may be used, depending on the designer's experience with similar applications and the degree of accuracy actually achieved. To use these values, the gearing must be maintained in accurate alignment and adequately lubricated. Spur gears must have properly designed profile modification and helical gears must have an axial contact ratio greater than 1.0. To incorporate this philosophy into the computer program, the decision was made to set $C_v = K_v = 0.9$ when $Q_v \geq 12$ and $v_t \geq 11.627$ m/sec.

If $v_t < 11.627$ m/sec and $Q_v \geq 12$ then the values of C_v and K_v ascribed to $Q_v = 11$ were followed, such that,

$$C_v = K_v = \left[\frac{92}{92 + \sqrt{200}\, v_t} \right]^{0.25} \qquad (6.11)$$

Whilst Clause 8.3.2 of AGMA 218.01 states that, based on experience, the values of C_v and K_v may be extrapolated beyond the end points of those shown in Figure 7 of AGMA 218.01, the decision was made to retain the defined end points in the computer program, as defined by equation (8.5M) of AGMA 218.01.

$$v_{t\ max} = [A + (Q_v - 3)]^2/200 \text{ m/s} \qquad (6.12)$$

However, if $Q_v < 6$, $v_{t\ max} = 13$ m/s, whilst if $Q_v > 12$, $v_{t\ max} = 66$ m/s.

Hence one of the checks that the program must perform, is to ensure that $v_{t\ max}$ has not been exceeded, for the nominated transmission accuracy level number, Q_v. In the event that $v_{t\ max}$ has been exceeded, the recommended Q_v is nominated.

6.5.2 Application factors (C_a) and (K_a)

The application factors make allowance for any externally applied loads in excess of the nominal tangential load, W_t. Application factors can only be established after considerable field experience is gained in a particular application.

In determining the application factors, consideration should be given to the fact that many prime movers develop momentary peak torques appreciably greater than those determined by the nominal ratings of either the prime mover or the driven equipment. There are many possible sources of overload which should be considered. Some of these are: system vibrations; acceleration torques; overspeeds in system operation; and changes in process load conditions.

6.5.3 Surface condition factor (C_f)

The surface condition factor C_f, used only in the pitting resistance formula, depends on:

i. surface finish as affected by cutting, shaving, lapping, grinding, shot peening, etc;
ii. residual stress; and
iii. plasticity effects.

Standard surface condition factors for gear teeth have not yet been established by AGMA for cases where there is a detrimental surface finish effect. In such cases, some surface finish factor greater than unity should be used.

6.5.4 Size factors (C_s) and (K_s)

The size factors reflect non-uniformity in material properties. They depend primarily on:

i. tooth size;
ii. diameter of parts;
iii. ratio of tooth size to diameter of part;
iv. facewidth;
v. area of stress pattern;
vi. ratio of case depth to tooth size; and
vii. hardenability and heat treatment of materials.

The size factor may be taken as unity for most gears, provided a proper choice of steel is made for the size of the part and its heat treatment and hardening process.

AGMA have not yet established standard size factors for gear teeth for cases where there is a detrimental size effect. In such cases, some size factor greater than unity should be used.

6.5.5 Load distribution factors (C_m) and (K_m)

The load distribution factors modify the rating equations to reflect the non-uniform distribution of load along the lines of contact. The amount of non-uniformity of the load distribution is caused by, and is dependent on, the following influences:

i. gear tooth accuracy, ie lead, profile, and spacing;
ii. alignment of the axis of rotation of the pitch cylinders of the mating elements;
iii. elastic deformations of gear elements, shafts, bearings and housings, and foundations which support the gear elements;
iv. bearing clearances;
v. Hertzian contact and bending deformations of the tooth surface;
vi. thermal expansion and distortion due to operating temperature (especially for wide faced gearing);
vii. centrifugal deflections due to operating speed; and
viii. tooth crowning and end relief.

The load distribution factor is defined by equation (13.1) of AGMA 218.01 as

$$C_m = K_m = C_{mf}\ C_{mt} \qquad (6.13)$$

where C_{mf} = face load distribution factor
C_{mt} = transverse load distribution factor

The transverse load distribution factor (C_{mt}), accounts for the non-uniform distribution of load among the gear teeth which share the total load.

AGMA have not yet established standardised procedures for the calculation of C_{mt}. A value of unity is currently recommended by AGMA.

The face load distribution factor (C_{mf}), accounts for the non-uniform distribution of load across the gearing facewidth. The magnitude of the face load distribution factor is defined as the peak load intensity divided by the average load intensity across the facewidth.

AGMA states that two methods may be used. One method is based on an analytical approach, while the other method is based on an empirical approach.

The analytical approach is based on the elastic and non-elastic lead mismatch. This method requires knowledge of the design, manufacturing, and mounting to evaluate the load distribution factor. The calculation of C_{mt} by this method has not been considered in this Thesis.

The empirical method requires a minimum amount of information. This method is recommended for normal, relatively stiff gear designs which meet the following requirements:

i. net facewidth to pinion pitch diameter ratios less than or equal to 2.0 (for double helical gears the gap is not included in the facewidth);

ii. the gear elements are mounted between bearings (not overhung);

iii. facewidth up to 1016 mm; and

iv. contact across full facewidth of narrowest number when loaded.

For normal, relatively stiff designs having gears mounted between bearings (not overhung) and relatively free from externally caused deflections, the following approximate method as given by equation (13.2) of AGMA 218.01, may be used.

$$C_{mf} = 1.0 + C_{mc} \, (C_{pf} \, C_{pm} + C_{ma} \, C_e) \tag{6.14}$$

where C_{mc} = lead correction factor
C_{pf} = pinion proportion factor
C_{pm} = pinion proportion modifier
C_{ma} = mesh alignment factor
C_e = mesh alignment correction factor

The lead correction factor (C_{mc}), modifies the peak load intensity when crowning or lead modification is applied.

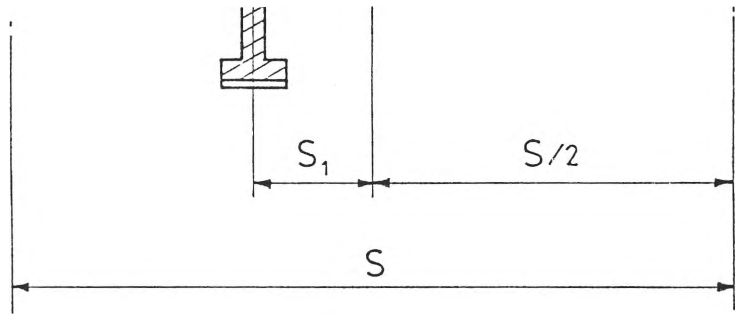
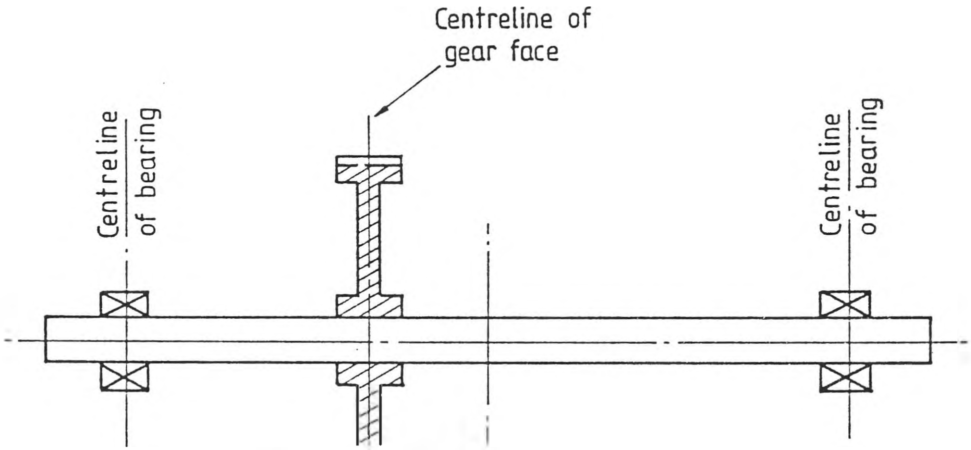


FIGURE 6.1 EVALUATION OF S AND S_1


Centreline of
gear face
Centreline
of bearing
Centreline
of bearing

For gears with unmodified leads AGMA recommends $C_{mc} = 1.0$, whilst for gears with leads properly modified by crowning or lead correction, AGMA recommends $C_{mc} = 0.8$.

The pinion proportion factor (C_{pf}), accounts for deflections due to load. The deflections are normally higher for wide facewidths or higher F/d ratios. The pinion proportion factor can be obtained from equations (13.3), (13.4M) and (13.5M) of AGMA 218.01, which are

When $F \leq 25.4$,

$$C_{pf} = F/10d - 0.025 \tag{6.15}$$

When $25.4 < F \leq 431.8$,

$$C_{pf} = F/10d - 0.0375 + 0.000492F \tag{6.16}$$

When $431.8 < F \leq 1016$,

$$C_{pf} = F/10d - 0.1109 + 0.000815 - 0.000000353F^2 \tag{6.17}$$

Note: For F/10d less than 0.05, use 0.05 for this value in equations (6.15), (6.16) or (6.17).

The pinion proportion modifier C_{pm}, alters C_{pf}, based on the location of the pinion relative to its bearing centre line.

For straddle mounted pinions with $(S_1/S) < 0.175$, AGMA recommends $C_{pm} = 1.0$, whilst for straddle mounted pinions with $(S_1/S) \geq 0.175$, AGMA recommends $C_{pm} = 1.1$ (refer Figure 6.1).

where S_1 = the offset of the pinion, ie, the distance from the bearing span centreline to the pinion mid face, mm.

S = the bearing span, ie, the distance between the bearing centre lines, mm.

The mesh alignment factor (C_{ma}), accounts for the misalignment of the axis of rotation of the pitch cylinders of the mating gear elements from all causes other than elastic deformations. The value of the mesh alignment factor can be obtained from equation (13.6) of AGMA 218.01, which is based on the accuracy of the gearing and misalignment effects which can be expected from four classes of gearing.

The values for the four classes are defined as follows.

$$C_{ma} = A + B(F) + C(F)^2 \qquad (6.18)$$

where A, B and C are empirical constants

Curve 1 (Open Gearing):

$A = 2.470 \times 10^{-1}$

$B = 0.657 \times 10^{-3}$

$C = -1.186 \times 10^{-7}$

Curve 2 (Commercial Enclosed Gear Units):

$A = 1.270 \times 10^{-1}$

$B = 0.622 \times 10^{-3}$

$C = -1.690 \times 10^{-7}$

Curve 3 (Precision Enclosed Gear Units):

$A = 0.675 \times 10^{-1}$

$B = 0.504 \times 10^{-3}$

$C = -1.440 \times 10^{-7}$

Curve 4 (Extra Precision Enclosed Gear Units):

$A = 0.380 \times 10^{-1}$

$B = 0.402 \times 10^{-3}$

$C = -1.270 \times 10^{-7}$

The mesh alignment correction factor (C_e), is used to modify the mesh alignment factor when manufacturing or assembly techniques improve the effective mesh alignment.

When the gearing is adjusted at assembly, or when the compatability of the gearing is improved by lapping, AGMA recommends $C_e = 0.8$. For all other conditions, AGMA recommends $C_e = 1.0$.

6.5.6 Hardness ratio factor (C_H)

The hardness ratio factor, C_H depends on:

i. gear ratio; and
ii. hardness of pinion and gear.

Values for C_H are applied to the wheel only, and not to the pinion.

For through hardened gears, where the pinion is substantially harder than the gear, the work hardening effect increases the gear capacity.

The C_H factor can be calculated from equation (15.1) of AGMA 218.01 as follows.

$$C_H = 1.0 + A\,(m_G - 1.0) \tag{6.19}$$

where $$A = 0.00898\,(H_{BP}/H_{BG}) - 0.00829 \tag{6.20}$$

H_{BG} = wheel Brinell hardness number (10 mm ball @ 3000 kg load)

H_{BP} = pinion Brinell hardness number (10 mm ball @ 3000 kg load)

Equation (6.20) is valid below a ratio $H_{BP}/H_{BG} = 1.7$. If H_{BP}/H_{BG} is less than 1.2, AGMA recommends that $C_H = 1.0$.

For surface hardened/through hardened gears, where surface hardened pinions (48 HRC or harder) are run with through hardened gears (180 to 400 BHN) a work hardening effect is achieved.

The C_H factor varies with the surface finish of the pinion, f_P, and the mating gear hardness.

The C_H factor can be calculated from equation (15.3) of AGMA 218.01 as follows.

$$C_H = 1.0 + B\ (450 - H_{BG}) \qquad (6.21)$$

where $B = 0.0075\ (e)^{-0.0112(f_P)}$ (6.22)

e = base of natural logarithms

f_P = surface finish of pinion (rms)

6.5.7 Life factors (C_L) and (K_L)

The life factors, C_L and K_L adjust the allowable stress numbers for the required number of cycles of operation.

The number of load cycles, N, is defined as the number of mesh contacts, under load, of the gear tooth being analysed. AGMA allowable stress numbers are established for 10^7 tooth load cycles. The life factor adjusts the stress numbers for design lives other than 10^7 cycles.

A C_L or K_L value of unity may be used, where justified by experience beyond 10 million cycles.

AGMA states that at present, there is insufficient data to provide accurate life curves for all types of gears and their applications. Experience, however, suggests that life curves for pitting and strength of steel gears are as shown in Figures 20 and 21 of AGMA 218.01, which have been reproduced as Figures 6.2 and 6.3 respectively.

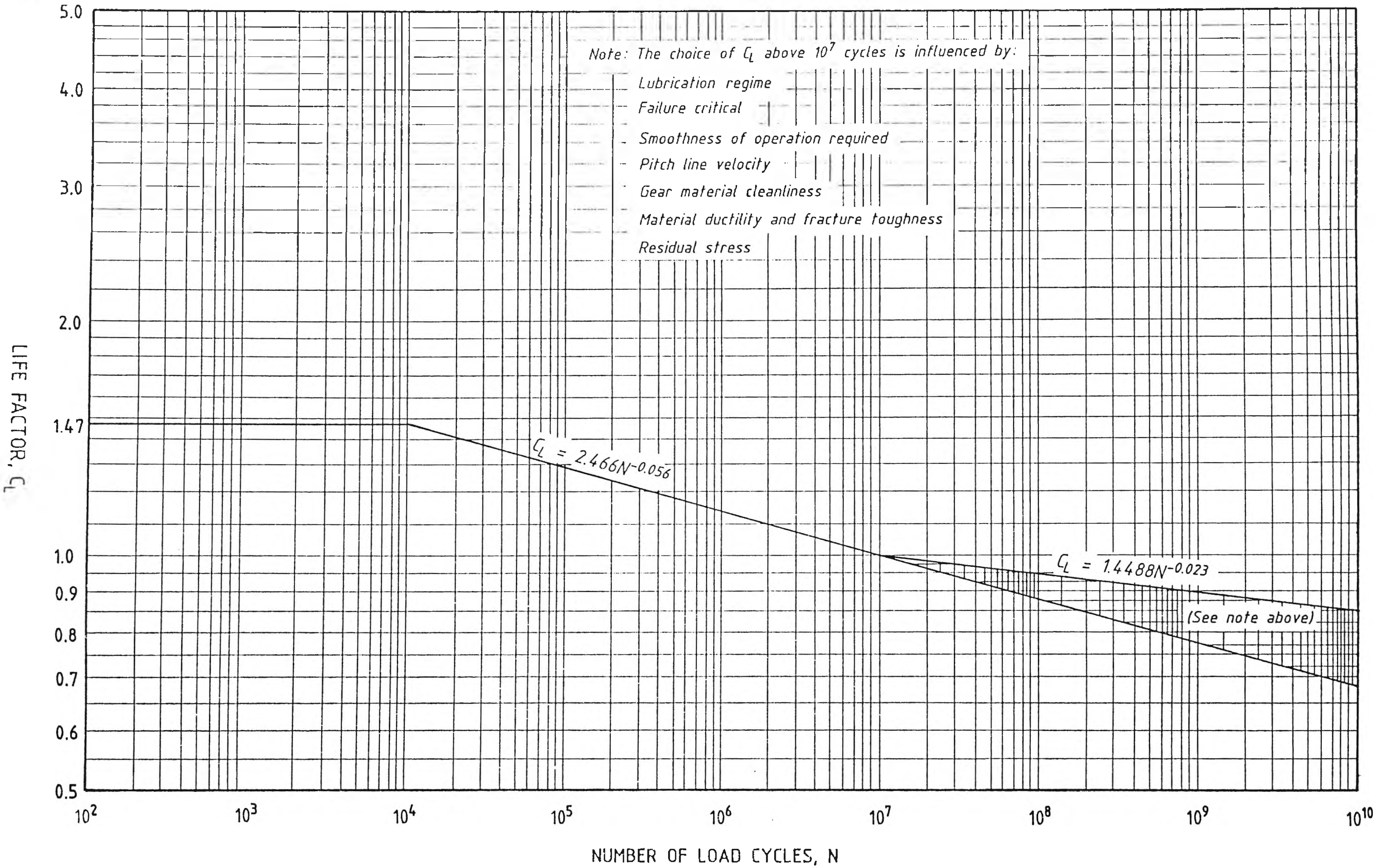


FIG. 6.2 PITTING RESISTANCE LIFE FACTOR, C_L

The equations for the pitting resistance life factor, C_L, can be obtained from Figure 6.2 as follows,

If $\quad N \leq 10,000$

$$C_L = 2.466\ (10,000)^{-0.056}$$

ie $$C_L = 1.47229 \tag{6.23}$$

If $\quad 10,000 < N \leq 10,000,000$

$$C_L = 2.466\ N^{-0.056} \tag{6.24}$$

If N > 10,000,000, then the value of C_L is influenced by the service to which the gears will be applied.

The upper portion of the shaded zone on Figure 6.2 is used for general commercial applications, whilst the lower portion is used for critical service applications, where little pitting and tooth wear is permissible, and where smoothness of operation and low vibration levels are required. To enable this concept to be incorporated into the computer program, the upper boundary was arbitrarily set at $Q_v \leq 6$, whilst the lower boundary was set at $Q_v \geq 12$, the distribution throughout the shaded area, was then considered to be linear.

The relationship between C_L and N, when plotted on log-log scales as shown by Figure 6.2, is linear and can be expressed by:

$$C_L = \text{Constant} * N^{\text{power}} \tag{6.25}$$

$$\text{where power} = \frac{\log C_{L2} - \log C_{L1}}{\log N_2 - \log N_1} \tag{6.26}$$

Considering the value of C_{L2}, when $N = 10^{10}$, then the upper boundary is defined by $C_{L2} = 0.8531$, corresponding to $Q_v \leq 6$, whilst for the lower boundary, $C_{L2} = 0.6792$, corresponding to $Q_v \geq 12$.

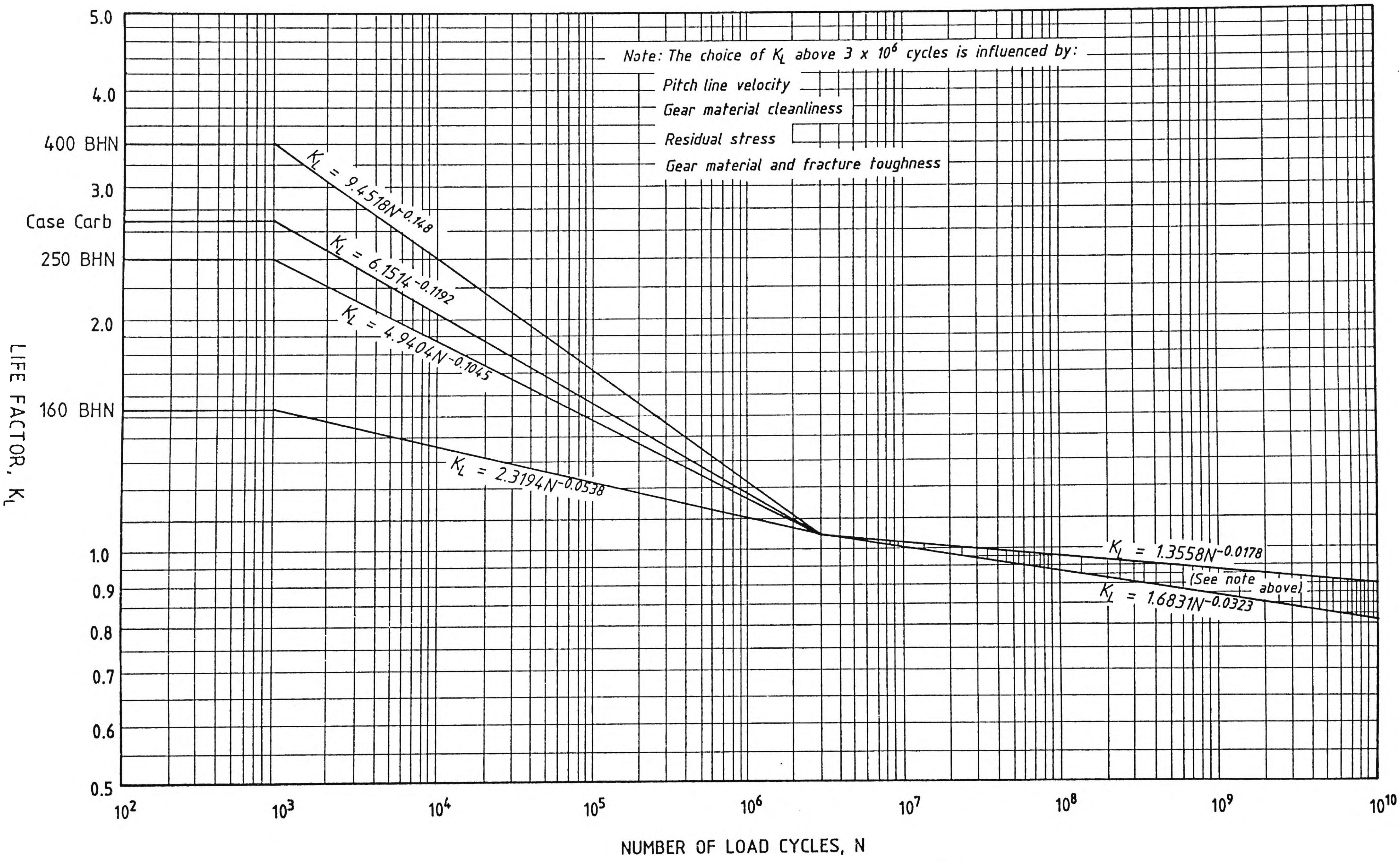


FIG. 6.3 BENDING STRENGTH LIFE FACTOR, K_L

On the arbitrary assumption that the relationship between C_{L2} and Q_v is linear, then,

$$C_{L2} = 1.027 - 0.029\ Q_v \tag{6.27}$$

Similarly, when $N = 10^7$, $C_{L1} = 1.0$.

Hence, the procedure to find C_L, when N > 10,000,000 is as follows,

If $Q_v \leq 6$,

$$C_L = 1.4488\ N^{-0.023} \tag{6.28}$$

If $Q_v < 12$,

$$C_{L2} = 1.027 - 0.029\ Q_v \tag{6.27}$$

and from equations (6.25) and (6.26),

$$\text{power} = (\log C_{L2})/3$$

$$C_L = (N/10^7)^{\text{power}} \tag{6.29}$$

If $Q_v \geq 12$,

$$C_L = 2.466\ N^{-0.056} \tag{6.24}$$

The equations for the bending strength life factor, K_L, can be obtained from Figure 6.3 as follows,

If $N < 3.0 \times 10^6$, then K_L is a function of the material hardness (BHN). Considering the value of K_L, when $N = 10^3$, then for BHN = 400, $K_L = 3.4$, whilst for BHN = 160, $K_L = 1.5995$. On the arbitrary assumption that the relationship between C_L and BHN is a second order polynomial, then when $N = 10^3$,

$$K_L = 0.011\ \text{BHN} - 0.00000625\ \text{BHN}^2 \tag{6.30}$$

The hardness of the gears above 400 BHN, presented some dilemma, as Figure 6.3 indicates that for case carburised gears (55-60 HRC), K_L is less than that for gears of 400 BHN.

Hence, an arbitrary decision was made, that if the BHN of a gear was greater than 400 BHN, then the value of K_L would be assigned based on a parabolic distribution of K_L, between the values of $K_L = 3.4$ at 400 BHN and $K_L = 2.7$ at 55-60 Rockwell "C" (≃ 600 BHN), when $N = 10^3$. The approximate relationship (19) between BHN and HRC, utilised in the program is,

$$BHN = [25{,}000 - 10\ (57 - HRC)^2]/[100 - HRC] \qquad (6.31)$$

Hence the procedure to find K_L is as follows.

If $N \leq 3{,}000{,}000$ and $BHN \leq 400$,

$$K_{L1} = 0.011\ BHN - 0.00000625\ BHN^2 \qquad (6.30)$$

else $N \leq 3{,}000{,}000$ and $BHN > 400$,

$$K_{L1} = 0.0165\ BHN - 0.00002\ BHN^2 \qquad (6.32)$$

If $N \leq 1{,}000$,

$$K_L = K_{L1}$$

Now when $N = 3{,}000{,}000$, $K_L = 1.0397$, for all conditions. Hence from equations (6.25) and (6.26), for $1{,}000 < N \leq 3{,}000{,}000$,

$$power = 0.00486 - 0.2876 \log K_{L1}$$

$$K_L = 1.0397\ [N/(3 \times 10^6)]^{power} \qquad (6.33)$$

It is of interest to note, that equations (6.30), (6.32) and (6.33) yield the following relationships, which may be compared to those of Figure 6.3.

<u>For BHN = 400,</u>

$$K_L = 9.4510\ N^{-0.1480} \quad \text{compared with} \quad K_L = 9.4518\ N^{-0.1480} \qquad (6.34)$$

<u>For BHN = 250,</u>

$$K_L = 4.7850\ N^{-0.1024} \quad \text{compared with} \quad K_L = 4.9404\ N^{-0.1045} \qquad (6.35)$$

<u>For BHN = 160,</u>

$K_L = 2.3210\ N^{-0.0538}$ compared with $K_L = 2.3194\ N^{-0.0538}$ (6.36)

<u>For BHN = 600,</u>

$K_L = 6.1515\ N^{-0.1192}$ compared with $K_L = 6.1514\ N^{-0.1192}$ (6.37)

If $N > 3{,}000{,}000$, then the value of K_L is influenced by the service to which the gears will be applied, similar to that for the value of C_L. When $N = 10^{10}$, then the upper boundary is defined by $K_{L2} = 0.8999$, corresponding to an arbitrary selection of $Q_v \leq 6$, whilst for the lower boundary, $K_{L2} = 0.8000$, corresponding to $Q_v \geq 12$.

On the arbitrary assumption that the relationship between K_{L2} and Q_v is linear, then,

$$K_{L2} = 1.0 - 0.01667\ Q_v \qquad (6.38)$$

Similarly, when $N = 3 \times 10^6$, $K_L = 1.0397$. Hence the procedure to find K_L, when $N > 3{,}000{,}000$ is as follows.

If $Q_v \leq 6$,

$$K_L = 1.3558\ N^{-0.0178} \qquad (6.39)$$

If $Q_v < 12$,

$$K_{L2} = 1.0 - 0.01667\ Q_v \qquad (6.38)$$

and from equations (6.25) and (6.26),

$$\text{power} = 0.28386 \log K_{L2} - 0.0048 \qquad (6.40)$$

$$K_L = 1.0397\ [N/(3 \times 10^6)]^{\text{power}} \qquad (6.33)$$

If $Q_v \geq 12$,

$$K_L = 1.6831\ N^{-0.0323} \qquad (6.41)$$

6.5.8 Reliability Factors (C_R) and (K_R)

The reliability factors account for the effect of normal statistical distribution of failures found in materials testing.

The allowable stress numbers given in Tables 5 and 6 of AGMA 218.01, are based on a statistical probability of one failure in 100 at 10^7 cycles. Table 6.1 contains reliability factors which may be used to modify these allowable stresses to change that probability.

Tooth breakages are sometimes considered to be a greater hazard than pitting. In such cases a value of K_R greater than C_R is selected.

Requirements of Application	C_R, K_R
Fewer than one failure in 10,000	1.50
Fewer than one failure in 1,000	1.25
Fewer than one failure in 100	1.00
Fewer than one failure in 10	0.85

TABLE 6.1 - RELIABILITY FACTORS, C_R and K_R

6.5.9 Temperature Factors C_T and K_T

The temperature factors C_T and K_T are generally taken as unity when gears operate with temperatures of oil or gear blank not exceeding 120^oC.

When gears operate at oil or gear blank temperatures above 120^oC, $C_T = K_T$ is given a value greater than one to allow for the effect of temperature on oil film and material properties.

Consideration must be given to the loss of hardness and strength of some materials due to the tempering effect of temperatures over 175^oC.

6.6 ELASTIC COEFFICIENT (C_p)

The elastic coefficient, C_p, is defined by equation (10.1) of AGMA 218.01, as

$$C_p = [\pi \{(1.0 - \mu_P^2)/E_P + (1.0 - \mu_G^2)/E_G \}]^{-0.5} \qquad (6.42)$$

where C_p = elastic coefficient, $(MPa)^{0.5}$
μ_P = Poisson's ratio for pinion = 0.3
μ_G = Poisson's ratio for wheel = 0.3
E_P = modulus of elasticity for pinion, MPa
E_G = modulus of elasticity for wheel, MPa

AGMA 218.01 recommends the following values for the modulus of elasticity. However, when more exact values for the modulus of elasticity are obtained from roller contact tests, they may be used.

Steel	Malleable Iron	Nodular Iron	Cast Iron	Aluminium Bronze	Tin Bronze
$2x10^5$	$1.7x10^5$	$1.7x10^5$	$1.5x10^5$	$1.2x10^5$	$1.1x10^5$

TABLE 6.2 - MODULUS OF ELASTICITY, E, (MPa)

6.7 Allowable Stress Numbers (s_{ac}) and (s_{at})

The allowable stress numbers depend on:

i. material composition;
ii. mechanical properties;
iii. residual stress; and
iv. hardness.

Allowable stress numbers for unity application factor and 10 million cycles of load application are determined or established from laboratory and field experience for each material and condition of that material. These stress numbers are designated s_{ac} and s_{at}. The allowable stress numbers are adjusted for design life load cycles by the use of the life factors, which have been discussed in Section 6.5.7.

6.7.1 Allowable contact stress number (s_{ac})

Allowable contact stress numbers for commonly used gear materials are shown in Table 5 of AGMA 218.01. Several values of s_{ac}, pertaining to through hardened and tempered steel gears, have been extracted from Table 5, and plotted as Figure 14 of AGMA 218.01. Both Table 5 and Figure 14 show a range of values for a particular material, and it is suggested that the lower value should be used for general design purposes.

In general, AGMA 218.01 has proven to be a most useful method for the rating of spur and helical gears. However, Tables 5 and 6 have proven to be somewhat of a disappointment, due mainly to the ambiguous way in which the gear materials have been specified. Presumably, the selection of materials for gears can be made from AGMA 390.03, Part II, Section 2. The type of material having been selected, AGMA 240.01(20) can be used for a detailed specification. However, the designation numbers in AGMA 218.01, AGMA 390.03 and AGMA 240.01 are not correlated. Further, AGMA 240.01 has been withdrawn since work on this Thesis commenced, all of which leaves the designer in an invidious situation.

It is hoped that this problem will be addressed in future editions of AGMA 218.01, such that a tabulation similar to Appendix A of BS436-1940(21) will be incorporated.

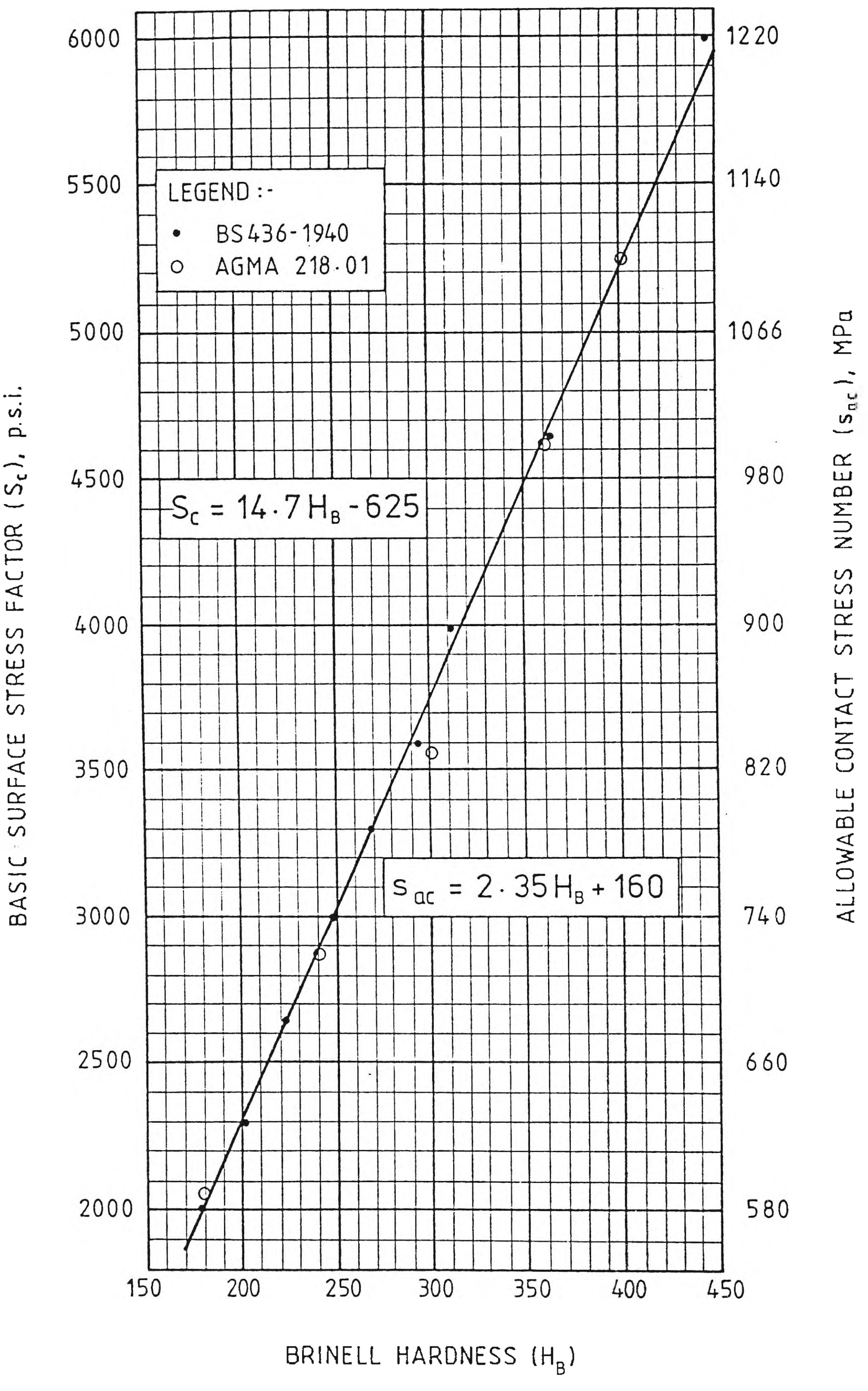


FIG. 6·4 APPARENT CORRELATION OF S_c AND s_{ac}

Material (BS970)	UTS (T/in^2)	BHN (BS240)	S_c (lb/in^2)	Correlation Factor	s_{ac} (MPa)
En8/Q	40	179	2000	$0.16S_c+260$	580
En14B/Q	40	179	2000	$0.16S_c+260$	580
En13	40	179	2000	$0.16S_c+260$	580
A1	-	180	2063	$6.25s_{ac}-1625$	590
En16/R	45	201	2300	$0.16S_c+260$	628
En17/R	45	201	2300	$0.16S_c+260$	628
En21/R	45	201	2300	$0.16S_c+260$	628
En9/S	50	223	2650	$0.16S_c+260$	684
En19/S	50	223	2650	$0.16S_c+260$	684
A2	-	240	2875	$6.25s_{ac}-1625$	720
En11/T	55	248	3000	$0.16S_c+260$	740
En18/T	55	248	3000	$0.16S_c+260$	740
En22/T	55	248	3000	$0.16S_c+260$	740
En25/T	55	248	3000	$0.16S_c+260$	740
En29/T	55	248	3000	$0.16S_c+260$	740
En19/U	60	269	3300	$0.16S_c+260$	788
En26/U	60	269	3300	$0.16S_c+260$	788
En11/V	65	293	3600	$0.16S_c+260$	836
En20/V	65	293	3600	$0.16S_c+260$	836
A3	-	300	3563	$6.25s_{ac}-1625$	830
En19/W	70	311	4000	$0.16S_c+260$	900
A4	-	360	4625	$6.25s_{ac}-1625$	1000
En28/Y	80	363	4650	$0.16S_c+260$	1004
A5	-	400	5250	$6.25s_{ac}-1625$	1100
En24/Z	100	444	6000	$0.16S_c+260$	1220
En30A	100	444	6000	$0.16S_c+260$	1220

TABLE 6.3 - APPARENT CORRELATION OF S_c AND s_{ac}

To illustrate the potential of a correlation between AGMA 218.01 and BS436-1940, the first five materials of Table 5 of AGMA 218.01 (which have euphemistically been referred to as hardened and tempered steel) have been tabulated against hardened and tempered forged steels from BS436, for a similar hardness range. This tabulation is shown as Table 6.3 (page 141), and is plotted as Figure 6.4.

Having plotted S_c values from Table 6.3, the line of best fit yields,

$$S_c = 14.7\ H_B - 625\ \text{(psi)} \qquad (6.43)$$

In assigning a suitable scale to s_{ac}, it can be seen that between 180 BHN and 400 BHN, S_c varies by 3230 units (5250-2020), whilst s_{ac} varies by 510 units (1100-590) over the same range. Hence, 500 S_c units will correspond to approximately 80 s_{ac} units on the same scale. Further, if there is to be a correlation, then a s_{ac} value of 590 MPa will correspond with a S_c value of approximately 2000 psi. This scaling procedure leads to a reasonable correlation as shown by Figure 6.4, yielding,

$$s_{ac} = 2.35\ H_B + 160\ \text{(MPa)} \qquad (6.44)$$

Combining equations (6.43) and (6.44) gives,

$$s_{ac} = 0.16\ S_c + 260\ \text{(MPa)} \qquad (6.45)$$

Hence, one may hypothesise that equation (6.45) represents a technique whereby detailed material specifications may be assigned to AGMA Class A1 to A5 gear materials. If there is validity to this hypothesis, then other materials which have been normalised, as opposed to hardened and tempered, within the forged steel category of BS 436 Appendix A, could presumably be utilised.

6.7.2 Allowable bending stress number (s_{at})

Allowable bending stress numbers for commonly used gear materials are shown in Table 6 of AGMA 218.01. Several values of s_{at}, pertaining to through hardened and tempered steel gears, have been extracted from Table 6, and plotted as Figure 15 of AGMA 218.01.

The comments made in Section 6.7.1, regarding the ambiguity of material specifications, are equally valid when using Table 6 of AGMA 218.01.

It should be noted, that when rating an idler gear, or other gears where the teeth are completely reverse loaded on every cycle, then the design value of s_{at} is to be 70 percent of the nominated value of s_{at} as found in Table 6 of AGMA 218.01.

6.8 An Example of the Use of the Computer Program

WOLLONGONG UNIVERSITY - SCHOOL OF MECHANICAL ENGINEERING
ANALYST: ROBERT DAVEY DATE: 9TH JAN 1989
THESIS EXAMPLE - SPUR GEARS
SPECIFIED DESIGN FACTORS OF AGMA 218.01 (DEC 1982).

SYMBOL	PINION	WHEEL	UNITS/ CLAUSE	DESCRIPTION
NP & NW	28	71		NUMBER OF GEAR TEETH
PSIS	0 00'00"	0 00'00"	DMS	HELIX ANGLE @ STANDARD PCD
F	52.000	52.000	MM	FACE WIDTH
MN	4.000	4.000	MM	NORMAL METRIC MODULE
X	.375	.143		ADDENDUM MODIFICATION COEFF
BN	.121	.156	MM	BACKLASH APPLIED
DLTARO	.000	.000	MM	TRUNCATION APPLIED
RPM	1440.000	567.887	RPM	REV PER MINUTE OF GEARS
LIFE	40.000	45.000	16.0	DESIGN LIFE OF GEAR(H*10**3)
SH	55.000	50.000	RC	SURFACE HARDNESS OF MAT.
SAC	1250.000	1200.000	MPA	CONTACT STRESS NUMBER
SAT	380.000	310.000	MPA	BENDING STRESS NUMBER
E	207.000	205.000	GPA	MODULUS OF ELASTICITY
MEW	.300	.300	10.0	POISSON'S RATIO
QV	8	6	8.3	QUALITY NUMBER FROM AGMA 390
CA	1.000	1.000	9.1	APPLIC. FCT. FOR PITTING RES
KA	1.200	1.200	9.1	APPLIC. FCT. FOR BENDING STR
CF	1.000	1.000	11.0	SURFACE CONDITION FACTOR
CS	1.000	1.000	12.2	SIZE FCT. FOR PITTING RES.
KS	1.000	1.000	12.2	SIZE FCT. FOR BENDING STR.
CMT	1.000	1.000	13.2.1	TRANSVERSE LOAD DISTRIB. FCT
CMC	1.000	1.000	13.2.2	LEAD CORRECTION FACTOR
CPM	1.100	1.100	13.2.2.1	PINION PROPORTIONAL MODIFIER
CE	1.000	1.000	13.2.2.1	MESH ALIGNMENT CORRECTION FC
CR	1.000	1.000	17.0	RELIABILITY FCT. PITTING RES
KR	1.000	1.000	17.0	RELIABILITY FCT. BENDING STR
CT	1.000	1.000	18.0	TEMPERATURE FCT. PITTING RES
KT	1.000	1.000	18.0	TEMPERATURE FCT. BENDING STR

RACK DIMENSIONS AND OPTIONS

SYMBOL	PINION	WHEEL	UNITS/ CLAUSE	DESCRIPTION
PHIC	20 00'00"	20 00'00"	DEG	NORMAL PRESSURE ANGLE
HA	5.000	5.000	MM	STANDARD ADDENDUM OF TOOL
HB	4.000	4.000	MM	STANDARD DEDENDUM OF TOOL
RT	1.520	1.520	MM	TIP RADIUS OF CUTTING TOOL
DELTAO	.000	.000	MM	PROTUBERANCE OF CUTTING TOOL
IDLER	2	1	14.4.1	NUMBER OF GEARS IN CONTACT
CURVE	2	2	FIG(10)	ALIGNMENT FACTOR CURVE
ROUGH	NO	NO		INACCURATE SPUR GEARS

WOLLONGONG UNIVERSITY - SCHOOL OF MECHANICAL ENGINEERING
ANALYST: ROBERT DAVEY DATE: 9TH JAN 1989
THESIS EXAMPLE - SPUR GEARS
GEAR RATING TO AGMA 218.01 (DEC. 1982).

SYMBOL	DESCRIPTION	PINION	WHEEL
N	NUMBER OF TEETH	28	71
RPM	SPEED OF ROTATION (REV/MIN)	1440.000	567.887
F	FACE WIDTH (MM)	52.000	52.000
PSIS	HELIX ANGLE (DMS)	0 00'00"	0 00'00"
MN	NORMAL MODULE (MM)	4.000	4.000
MT	TRANSVERSE MODULE (MM) MT=MN/COS(PSIS)	4.000	4.000
D	OPERATING PITCH CIRCLE DIAMETER (MM)	113.130	286.866
K10	F * RPMP * DP / (1.91*10**7)	.444	.444
IDLER	NUMBER OF GEARS IN CONTACT	2	1
I	GEOMETRY FACTOR FOR PITTING RESISTANCE	.120	.120
J	GEOMETRY FACTOR FOR BENDING STRENGTH	.479	.435
SH	SURFACE HARDNESS (RC)	55.000	50.000
VT	PI * RPMP * DP / 60000 (M/SEC)	8.530	8.530
QV	QUALITY NUMBER - AGMA 390	8	6
CV	FIG 7	.748	.648
KV	FIG 7	.748	.648
CA	CLAUSE 9.1	1.000	1.000
KA	CLAUSE 9.1	1.200	1.200
CP	TABLE 4 (MPA ** 0.5)	189.809	189.809
CF	CLAUSE 11.1	1.000	1.000
CS	CLAUSE 12.2	1.000	1.000
KS	CLAUSE 12.2	1.000	1.000
CMT	CLAUSE 13.2.1	1.000	1.000
CMC	CLAUSE 13.2.2.1	1.000	1.000
F/D	FACE WIDTH TO DIAMETER RATIO	.460	.181
CPF	FIG 8	.038	.038
CPM	CLAUSE 13.2.2.1	1.100	1.100
CMA	FIG 10 (CURVE 2)	.159	.159
CE	CLAUSE 13.2.2.1	1.000	1.000
CMF	1.0 + CMC * (CPF*CPM+CMA*CE)	1.201	1.201
CM	CMF * CMT	1.201	1.201
KM	CMF * CMT	1.201	1.201
SAC	TABLE 5 (MPA)	1250.000	1200.000
SAT	TABLE 6 (MPA)	380.000	310.000
CH	FIG 18	1.000	1.000
N	NUMBER OF LOAD CYCLES * 10**6	6912.000	1533.296
CL	FIG 20	.805	.891
KL	FIG 21	.874	.930
CR	TABLE 8	1.000	1.000
KR	TABLE 8	1.000	1.000
CT	CLAUSE 18.1	1.000	1.000
KT	CLAUSE 18.1	1.000	1.000
PITTING	2		
PAC	K10 * I * CV * DP * (SAC * CL * CH)	105.144	102.761
	----------------- -----------------		
(KW)	CS * CM * CF * CA (CP * CT * CR)		
BENDING			
PAT	K10 * J * KV * SAT * KL * MT	102.618	100.106

(KW)	KA * KS * KM * KR * KT		

SECTION 7

GEAR DESIGN TO AGMA 218.01

7.1 Introduction

The combination of the preceding sections has led to the development of computer software for the design of spur or helical gears.

In designing a set of gears, an engineer usually knows only three pieces of data, namely, the power to be transmitted, the required gear ratio and a particular speed of rotation. With this data as a starting point, gear design constitutes a difficult set of computations, due to the nature of the trial and error approach, long associated with gear design.

The software that has been developed will design a set of gears to suit the requirements of "the three pieces of data" approach, yet has sufficient flexibility to accommodate the most stringent requirements of an experienced gear designer.

The "internal intelligence" of the software has been developed over a number of years, but in the absence of other constraints, aims to design the gears on the basis of an equal slide to roll ratio. It is the author's firm conviction, based on both theoretical and more importantly numerous practical applications, that this approach yields the "best" set of gears for the majority of applications.

7.2 Slide to Roll Ratio for Spur Gears at Standard Centres

The slide to roll ratio at any point in the mesh of two gears is defined as the ratio of the instantaneous sliding velocity to the instantaneous rolling velocity at that point. Since both components of the ratio vary directly with angular velocity, the ratio is independent of this value and is in fact dimensionless.

The rolling velocity is defined as the velocity of the point of contact over the surface in question due solely to the rolling action. It is equal to the product of the instantaneous radius of curvature at the point of contact and the angular velocity. In any mesh there will be two different rolling velocities, one for each member, due to the non-proportional differences in instantaneous radius of curvature and angular velocity. This indicates that sliding must occur at a velocity equal to the algebraic difference between the two rolling velocities.

It is clear that there is no value of slide to roll ratio peculiar to either member in isolation; it can only exist in relation to a pair at some specified point in their mesh.

The points usually chosen are the extremes of the path of contact. These points are sometimes referred to as "at start/finish of mesh" or "approach/recess" or similar. Under this type of convention it is necessary to know which member is driving before the definition is clear. In cases where the members drive alternately it can become confusing. Another convention refers to the same points as "at pinion/wheel tip" and this is independent of driver/driven sense.

In general terms, a high slide to roll ratio indicates a high sliding velocity, and a low ratio indicates a slow sliding velocity. This may not be true when contact occurs close to the base circle of either member as in this zone gross changes occur in the instantaneous radius of curvature as contact moves away from it. However, because of the high Hertzian stresses associated with this zone, designers try to avoid this condition and in most cases the above statement is generally true.

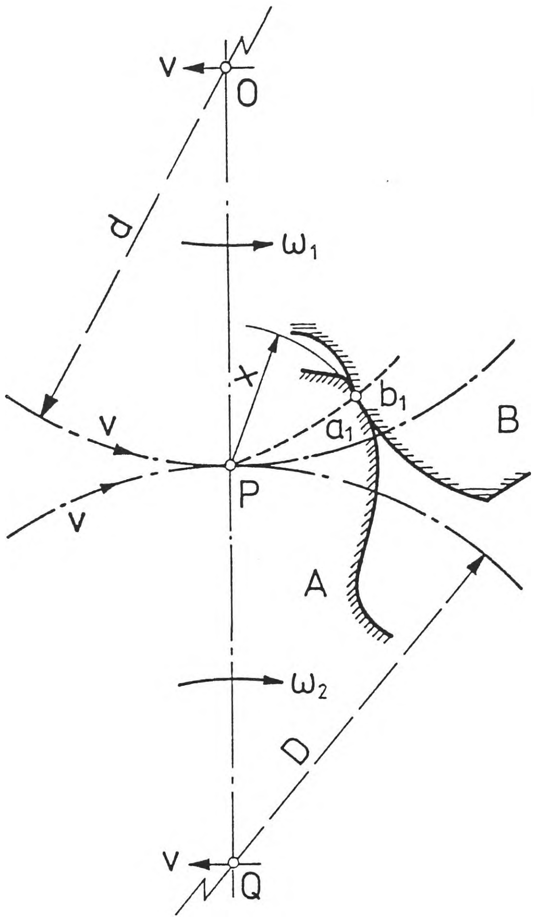


FIG. 7·1 VELOCITY OF SLIDING

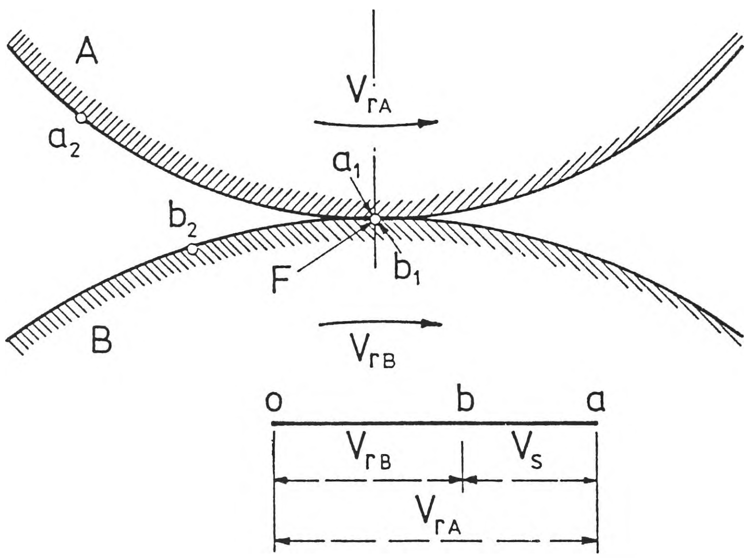


FIG. 7·2 SIMPLIFIED REPRESENTATION OF SLIDING AND ROLLING

High sliding velocities in a mesh indicate high friction losses and resulting inefficiency. It has been found that differences in efficiency due to this cause are not large, and in many cases are not significant. However, such lost energy shows up as heat, and in some high power installations, a significant improvement in running temperature can be achieved by minimising sliding velocities.

High sliding velocities can also magnify wear rates leading to earlier than necessary failures.

In most cases designers will try to minimise sliding in the mesh. If the slide to roll ratio is made the same at the beginning and the end of the mesh, it is considered to be "balanced", with sliding held to a minimum. In general, a designer will attempt to produce a design with a reasonable degree of balance in the slide to roll ratios at the extremes of the contact path.

In the case of very slow moving gears it may be advantageous to design for a high slide to roll ratio at the beginning of mesh to make the sliding velocity high, and hence enhance the formation of a lubricant wedge at the start of the mesh.

The slide to roll ratio at the extremes of the mesh is very sensitive to changes in diameter, the degree depending on many factors. It can be shown that even manufacturing tolerances may produce a wide range in the ratio. The designer's objective should be to achieve a general balance, within a tolerance range, rather than a precise mathematical balance. The following mathematical analysis will amplify the preceding comments, and should aid the designer in the selection of a balanced slide to roll ratio.

Suppose that the teeth A and B (shown in Figure 7.1) make contact at a point a_1b_1 distance x from the pitch point P. Now, the pitch circles at P may be brought to rest by giving an equal and opposite velocity to the centres O and Q. When this is done, P becomes the instantaneous centre of rotation of the line Pa_1, and the point a_1 on tooth A will then have a rolling velocity given by

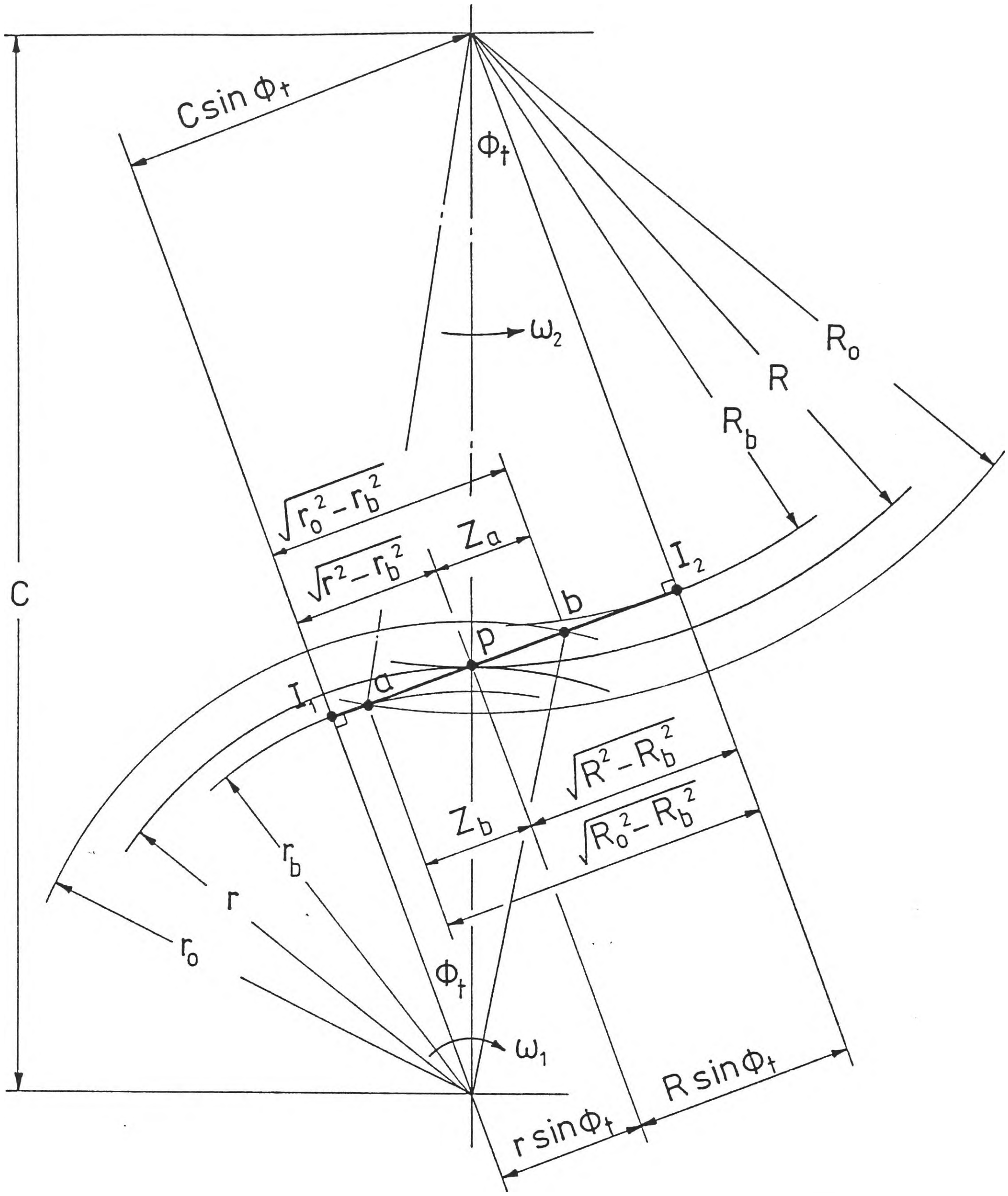


FIG. 7.3 COMMON NORMAL AT POINT OF CONTACT

$$V_{rA} = \frac{V.a_1P}{QP} = \frac{2\ v\ x}{D} = x\ \omega_2 \qquad (7.1)$$

in a direction at right angles to Pa_1, while the point b_1 on tooth B will have a rolling velocity given by

$$V_{rB} = \frac{2\ v\ x}{d} = x\ \omega_1 \qquad (7.2)$$

in the same line, but opposite sense.

The rolling velocity of a_1 relative to the rolling velocity of b_1 is the velocity of sliding between the teeth. Hence,

$$V_s = V_{rA} - V_{rB} \qquad (7.3)$$

A clearer picture of the relative motion of the profiles is shown in Figure 7.2. Here the profiles A and B are shown making contact at points a_1 and b_1 at the fixed point F; at a later instant both profiles will have moved in the direction of their length with the appropriate velocities of rolling, and points a_2 and b_2 will have arrived at F. When $a_1a_2 = b_1b_2$ "pure rolling" results.

Ideally, one would wish to design a gear set to minimise the amount of sliding between the gear teeth, with a greater emphasis on a rolling action. Experience has shown that this design concept enhances the life of the gears and facilitates smoother operation.

The following analysis shows the relative variation of sliding and rolling for different points of contact of involute teeth.

Figure 7.3 shows the common normal at the point of contact. If the pitch point P is considered to be the origin of a set of cartesian co-ordinates, which encompassses the maximum possible path of contact I_1I_2, then Figure 7.4 can be drawn, to aid in the formulation of equations, for the calculation of the ratio of the sliding to the rolling velocity, at the point of contact, which is sometimes termed "specific sliding".

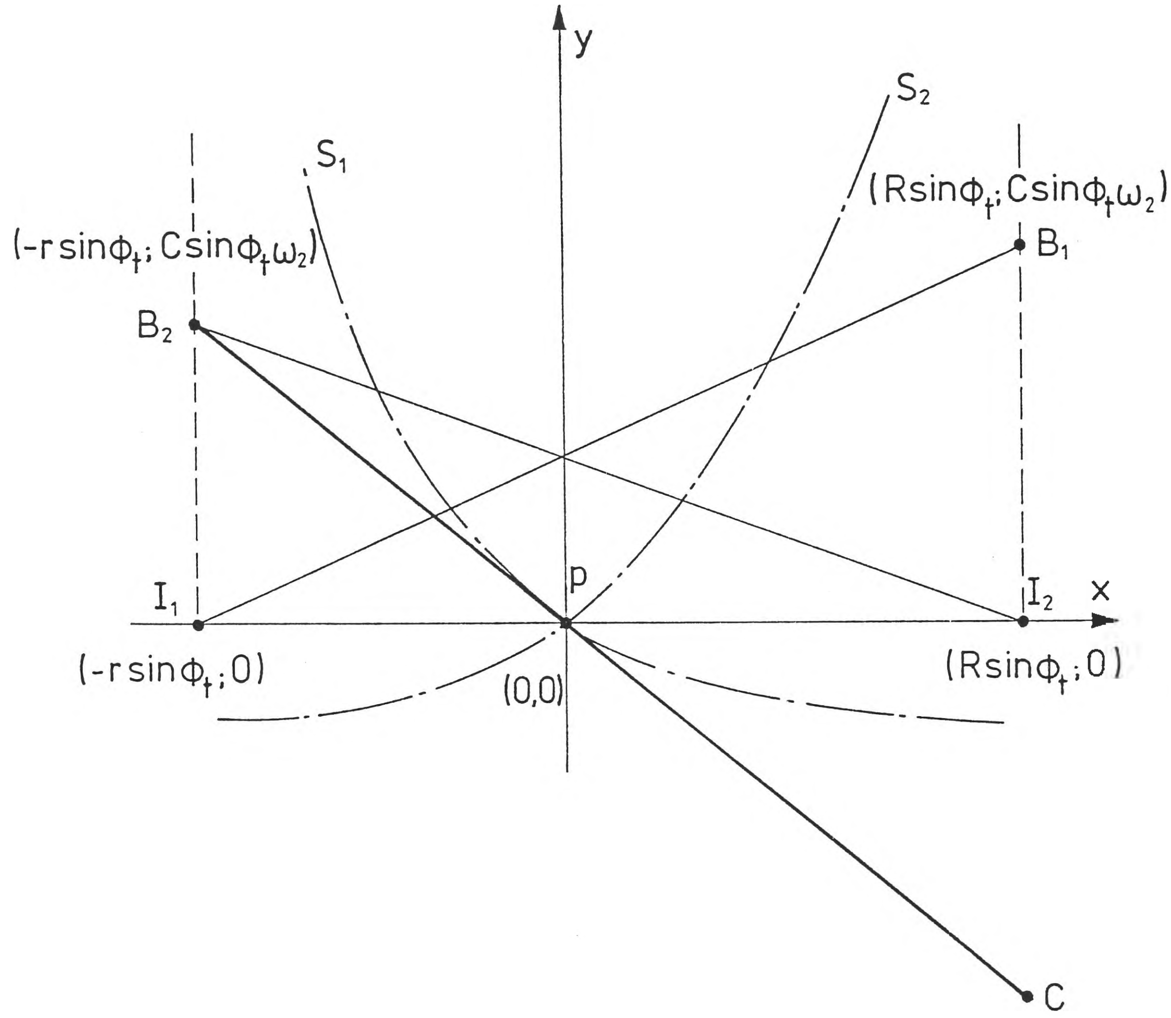


FIG. 7.4 SPECIFIC SLIDING EQUATIONS

Considering Figure 7.4, the equations are as follows.

The line I_1B_1 represents the rolling velocity of point B.

$$\frac{y - 0}{x + r\sin(\phi_t)} = \frac{C\sin(\phi_t)\,\omega_1 - 0}{R\sin(\phi_t) + r\sin(\phi_t)} = \omega_1$$

$$y = x\,\omega_1 + \omega_1\,r\sin(\phi_t) \qquad (7.4)$$

$$y = m_G\,\omega_2\,x + \omega_2\,R\sin(\phi_t) \qquad (7.5)$$

The line B_2I_2 represents the rolling velocity of point A.

$$\frac{y - C\sin(\phi_t)\,\omega_2}{x + r\sin(\phi_t)} = \frac{0 - C\sin(\phi_t)\,\omega_2}{R\sin(\phi_t) + r\sin(\phi_t)} = -\,\omega_2$$

$$y = -\,\omega_2\,x - \omega_2\,r\sin(\phi_t) + \omega_2\,C\sin(\phi_t)$$

$$y = -\,\omega_2\,x + \omega_2\,R\sin(\phi_t) \qquad (7.6)$$

From equation (7.3) the sliding velocity $B_2C = B_2I_2 - I_1B_1$

The line B_2C represents the sliding velocity of A relative to B.

$$y = -\,\omega_2\,x + \omega_2\,R\sin(\phi_t) - m_G\,\omega_2\,x - \omega_2\,R\sin(\phi_t)$$

$$y = -\,\omega_2\,x\,(1 + m_G) \qquad (7.7)$$

Similarly the specific sliding or slide to roll ratio is equal to either the ratio of $B_2C{:}I_1B_1$ or $B_2C{:}B_2I_2$.

The ratio of B_2C to I_1B_1 represents the first case of specific sliding.

$$\frac{B_2C}{I_1B_1} = \frac{-\,\omega_2\,x\,(1 + m_G)}{\omega_2\,m_G\,x + \omega_2\,R\sin(\phi_t)}$$

$$y = \frac{-\,(1 + m_G)\,x}{m_G\,x + R\sin(\phi_t)} \qquad (7.8)$$

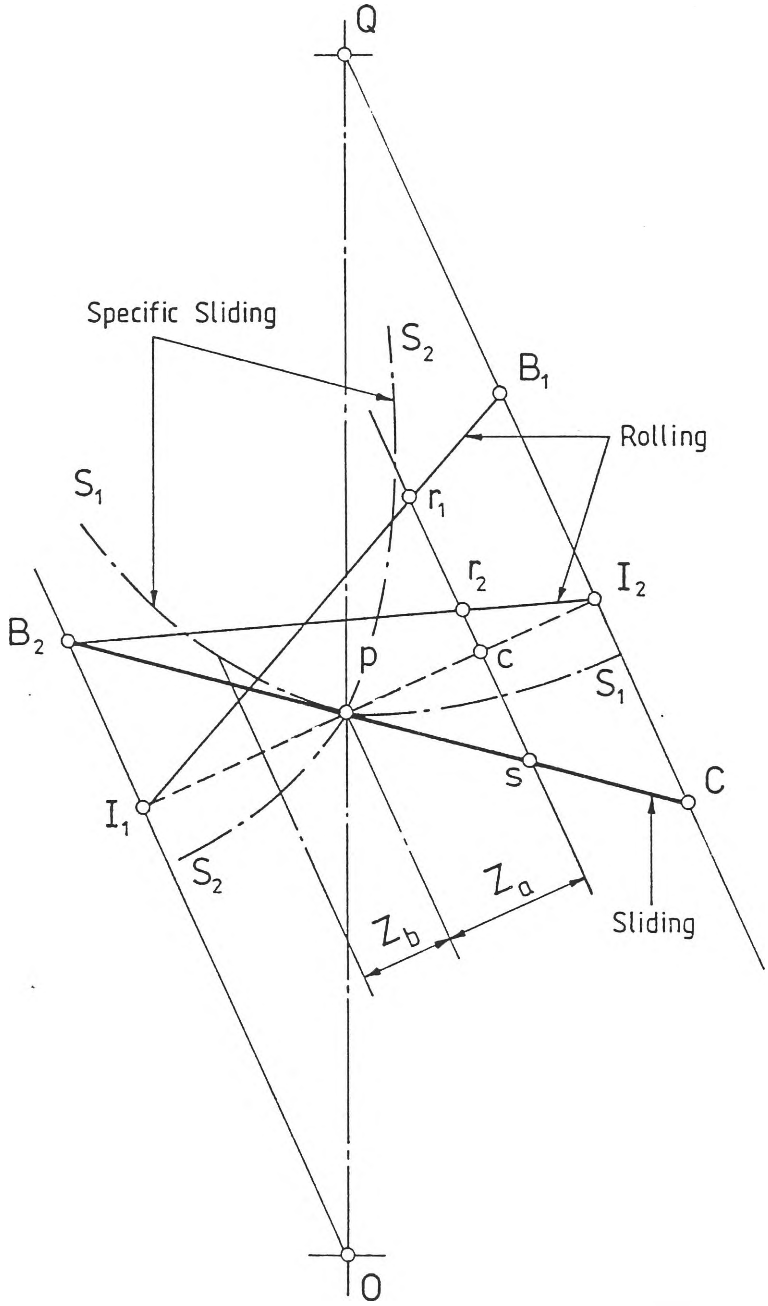


FIG. 7.5 ROLLING AND SLIDING VELOCITIES AND SPECIFIC SLIDING OF INVOLUTE TEETH

Similarly for the second case, with a change in the sign of B_2C,

$$\frac{B_2C}{B_2I_2} = \frac{\omega_2\ x\ (1 + m_G)}{-\ \omega_2\ x + \omega_2\ R\ \sin(\phi_t)}$$

$$y = \frac{(1 + m_G)\ x}{R\ \sin(\phi_t) - x} \qquad (7.9)$$

The curves S_1S_1 and S_2S_2 in Figure 7.5 represent the specific sliding. At the point c, for example, the value of the specific sliding for the gear on centre O is equal to cs/cr_1, and for the gear on centre Q it is equal to cs/cr_2, whilst I_1c and I_2c are the respective radii of curvature of the profiles at c. The value is unity when the point of contact reaches the interference point of the mating gear; for the mating gear itself the value is infinity. At the extremes of the approach and recess paths, Z_b and Z_a respectively, the specific sliding will be a maximum for a particular set of gears. Hence the technique in gear design is to distribute the addendum modification coefficients in such a way, that Z_b and Z_a cause the slide to roll ratios to be approximately equal. It is clear from inspection of Figure 7.5, that when the slide to roll ratios at the extremes of the path of contact are equal, they are also a minimum. From equations (7.8) and (7.9), the slide to roll ratios will be equal when $-Z_b$ is substituted for x and Z_a is substituted for x, providing that:

$$\frac{Z_b}{R\ \sin(\phi_t) - m_G\ Z_b} = \frac{Z_a}{R\ \sin(\phi_t) - Z_a}$$

Rearranging gives

$$\frac{Z_a - Z_b}{Z_a\ Z_b} = \frac{m_G - 1}{R\ \sin(\phi_t)} \qquad (7.10)$$

where $Z_a = [r_o^2 - r_b^2]^{0.5} - [r^2 - r_b^2]^{0.5}$

$Z_b = [R_o^2 - R_b^2]^{0.5} - [R^2 - R_b^2]^{0.5}$

and r_o, R_o, R and $\phi_t = f(x)$, where x is the addendum modification.

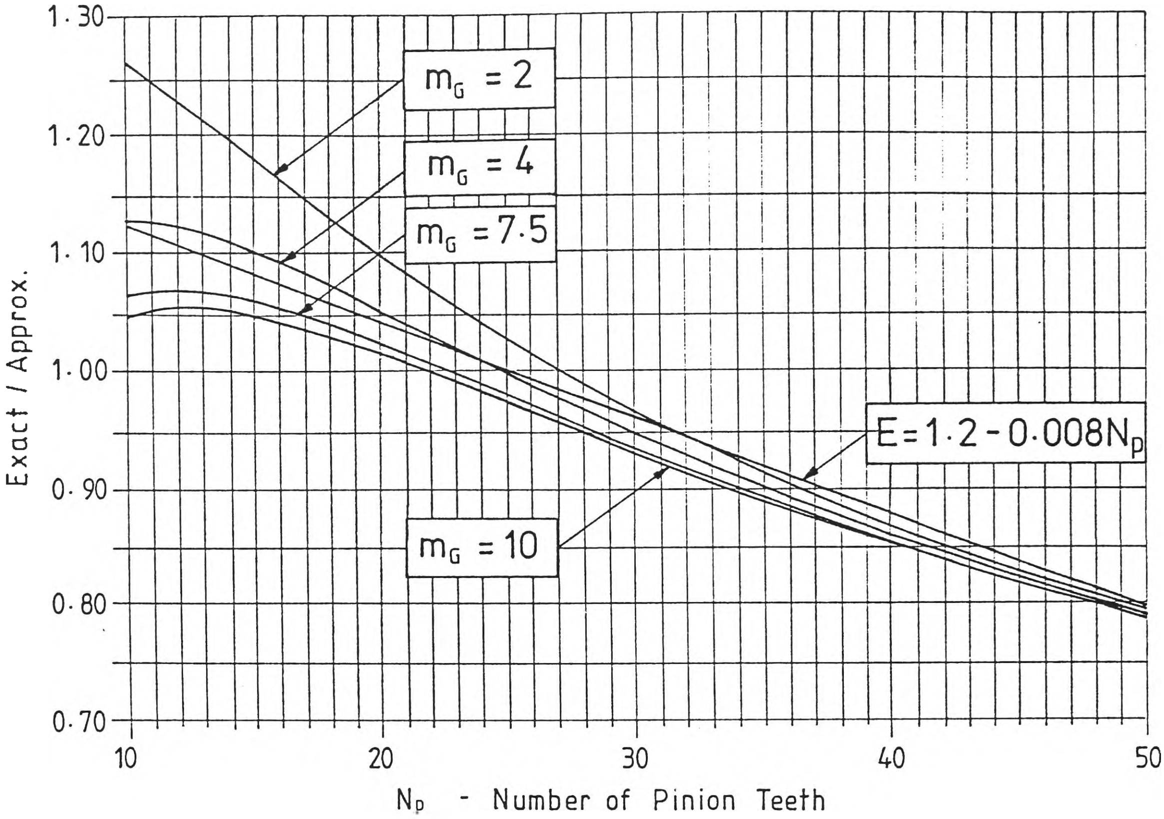


FIG. 7.6 BS PD6457 APPROXIMATION FOR EQUAL SLIDE TO ROLL

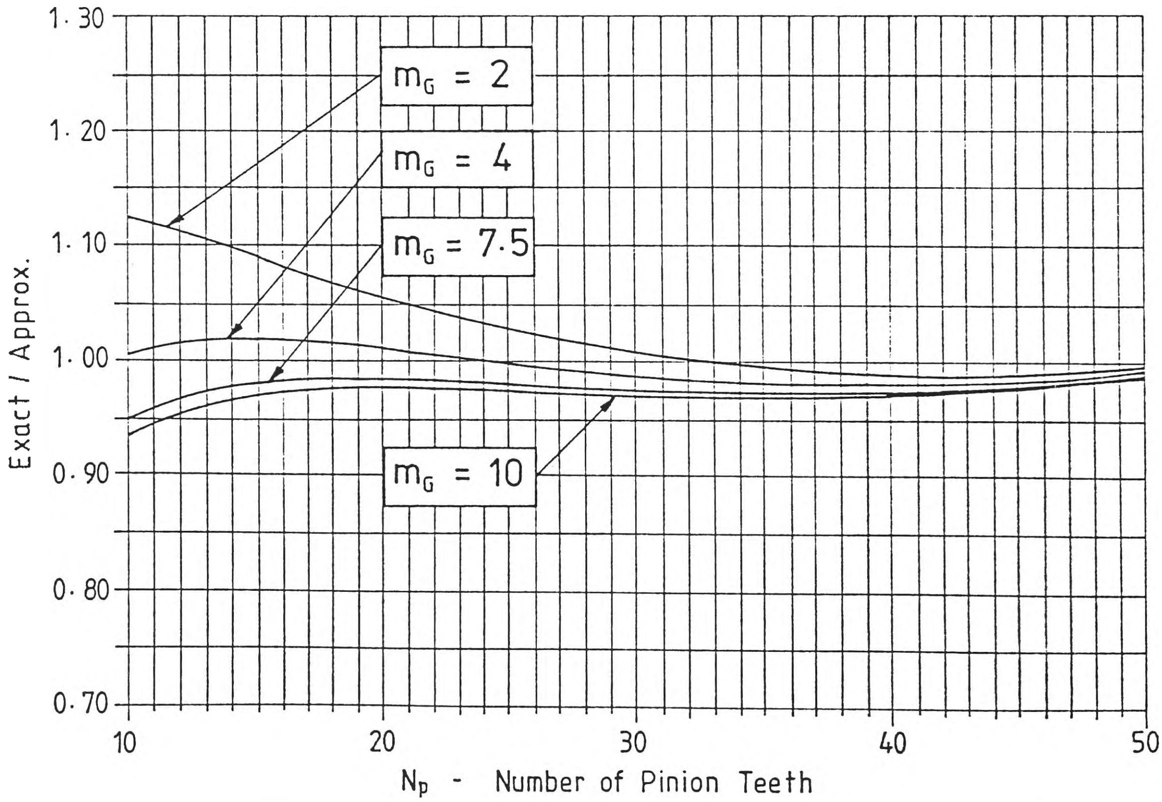


FIG. 7.7 DAVEY'S APPROXIMATION FOR EQUAL SLIDE TO ROLL

Equation (7.10) has proven to be insoluable in terms of making the addendum modification the subject of the equation. However, BSI PD6457 (15) has proven to be some source of encouragement in that Appendix A2, equation (2.3), suggests an approximate solution to equation (7.10).

Assuming unit module, then $R \sim 0.5\ m_G\ N_p$, the approximation being due to the fact that R is a function of the sum of the addendum modifications. Hence equation (7.10) becomes

$$\frac{Z_a - Z_b}{Z_a\ Z_b} = \frac{2\ (1 - 1/m_G)}{N_p\ \sin(\phi_t)}$$

Rearranging gives

$$\frac{\sqrt{N_p}\ (Z_a - Z_b)\ \sin(\phi_t)}{Z_a Z_b} = \frac{2}{\sqrt{N_p}}\left[1 - \frac{1}{m_G}\right] \tag{7.11}$$

The corresponding equation from BS PD6457 for the special case when $x_P = -x_G$ is:

$$x_P = \frac{2}{\sqrt{N_P}}\left[1 - \frac{1}{m_G}\right]$$

Hence if equation (7.10) is solved by an iteration process in a computer, then the ratio of the exact value for x_P to that of the BS PD6457 approximation is

$$\frac{\text{EXACT}}{\text{APPROX}} = \frac{x_P\ Z_a\ Z_b}{\sqrt{N_P}\ (Z_a - Z_b)\ \sin(\phi_t)} \tag{7.12}$$

where x_P is from equation (7.10).

Equation (7.12) is plotted as Figure 7.6, which shows that for pinion teeth numbers less than 25, the BS approximation is too small, whilst for pinion teeth numbers above 25, the BS approximation is too large.

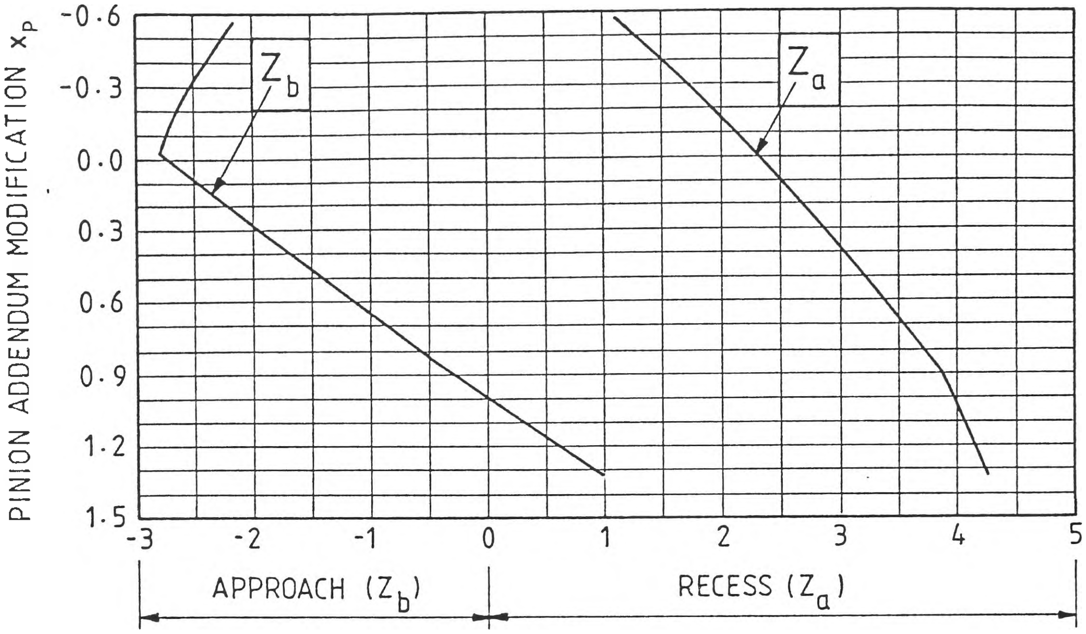


FIG. 7.8 THE PATH OF CONTACT AS A FUNCTION OF x_p

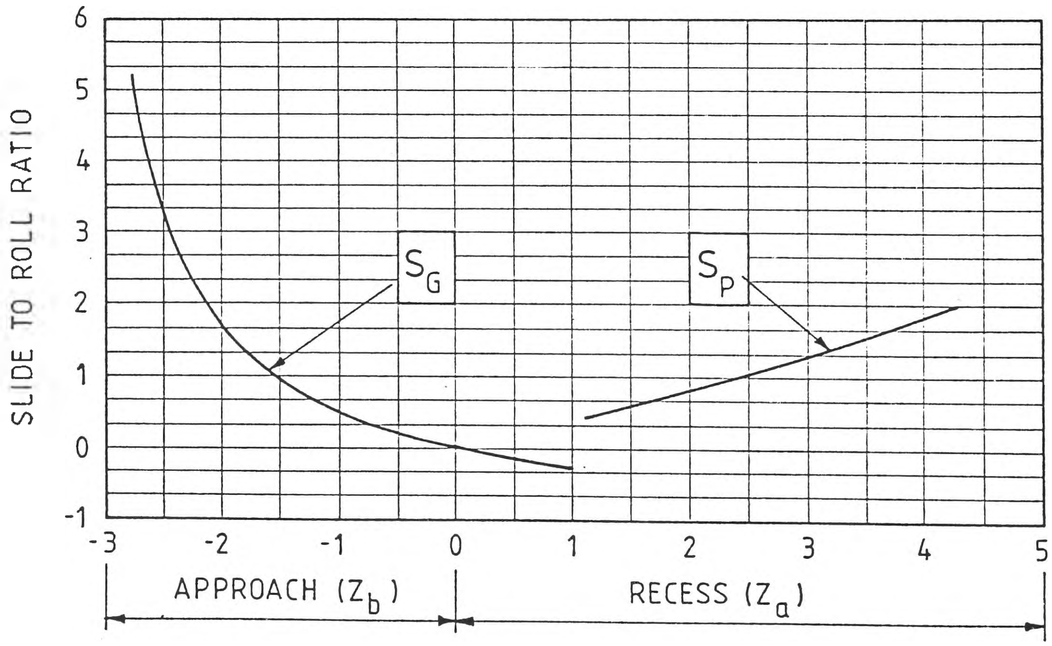


FIG. 7.9 THE SLIDE TO ROLL RATIO AS A FUNCTION OF Z

If one assumes that the error is approximately linear, varying from a value of 1.12 at $N_P = 10$ to 0.80 at $N_P = 50$, then the error can be expressed as

$$E = 1.2 - 0.008\ N_P \tag{7.13}$$

Hence a better approximation of BSI PD6457 equation (2.3) may be

$$x_P = \frac{2.4 - 0.016\ N_P}{\sqrt{N_P}} \left[1 - \frac{1}{m_G} \right] \tag{7.14}$$

Equation (7.14) is plotted as Figure 7.7. However, for all practical purposes, the formula advocated by BSI PD6457 may be utilised. If for some reason the exact solution is required, then equation (7.10) can be solved by an iteration process.

To give a clearer picture of the effect of addendum modification on Z_a and Z_b, and hence the slide to roll ratio, it is beneficial to analyse an example. Consider a spur gear set with an input speed of 1 rad/sec, having 20 pinion teeth and 100 wheel teeth. The gears are accurately manufactured, utilising a unit module ISO 53 cutter, and have addendum modifications such that $x_P = -x_G$, causing the gears to run at standard centres. Further, to ensure that only "real" gear sets were analysed, the following limitations were placed on the combinations being considered:

i. The involute clearance coefficient, C_I, was set at 0.02 (refer to Section 7.3).
ii. The bottom clearance coefficient, C_U, was set at 0.10.
iii. The tip width coefficient, C_W, was set at 0.20 (refer to Section 7.4).
iv. The total contact ratio, m_T, was set at 1.1.

Figure 7.8 shows the effect of addendum modification on the approach path, Z_b, and the recess path, Z_a. It should be noted that in this example, it is possible for the gears to operate with no approach path, when x_P exceeds +1.0, ie, when the outside diameter of the gear equals the standard pitch circle diameter. The slide to roll ratio as a function of the approach and recess paths is plotted as Figure 7.9.

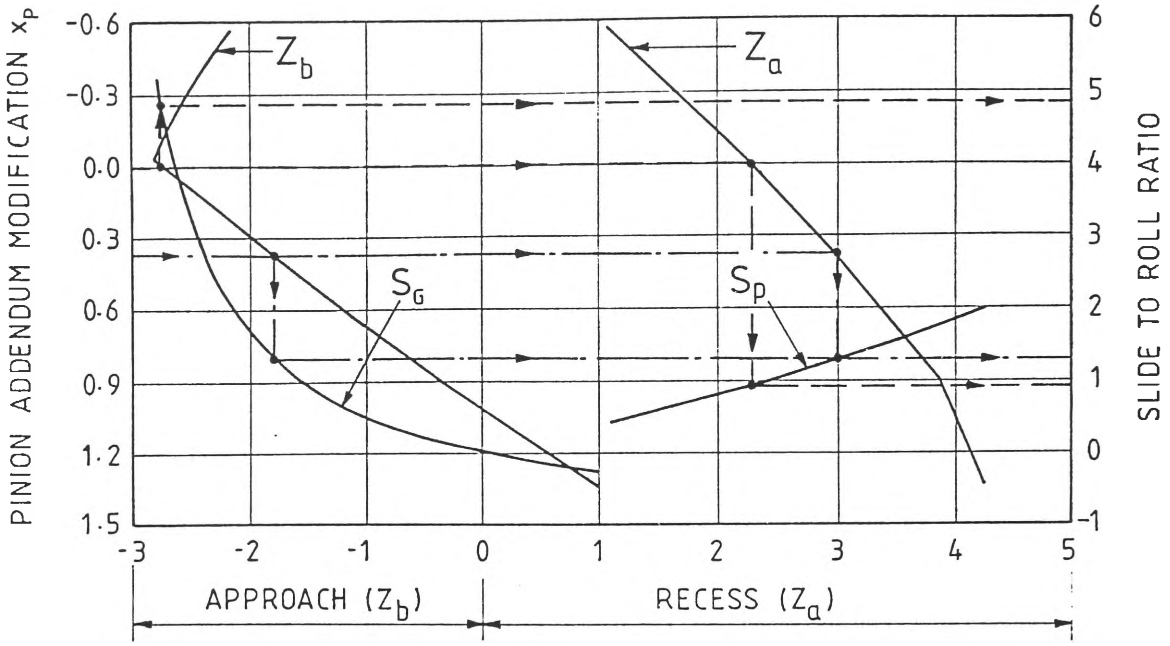


FIGURE 7.10 EQUAL SLIDE TO ROLL RATIO

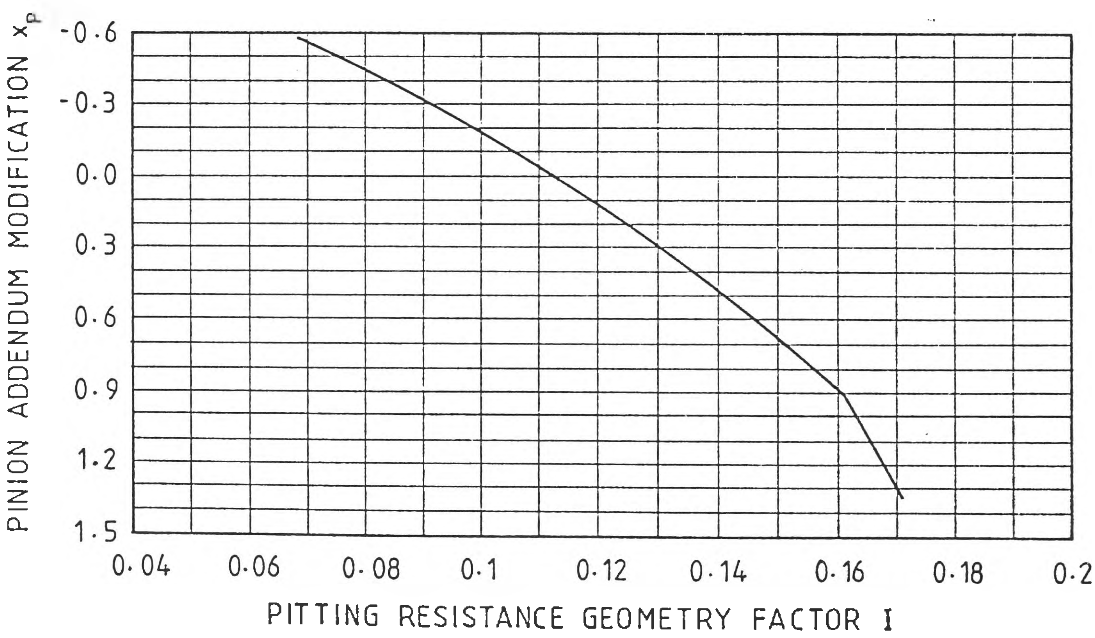


FIGURE 7.11 GEOMETRY FACTOR I AS A FUNCTION OF x_p

It is of interest to note that if the gears are manufactured with no addendum modification, ie $x_P = -x_G = 0.0$, then $Z_b = 2.734$ and $Z_a = 2.298$, corresponding to a slide to roll ratio on the pinion of 0.931 and on the wheel of 4.785. However, if $x_P = -x_G = 0.372$, then $Z_b = 1.758$ and $Z_a = 2.984$, corresponding to a slide to roll ratio of 1.268 on both the pinion and wheel. Figure 7.10 illustrates this particular point.

The slide to roll ratio has some effect on the scuffing and lubrication of the gears, which in turn can be reflected in the wear rate. Hence, as AGMA 218.01(4) rates gears as a function of the pitting resistance geometry factor, I, it would be prudent to examine I as a function of addendum modification, which by implication is a function of the approach path, Z_b, and the recess path, Z_a. The relationship of I as a function of addendum modification is shown in Figure 7.11.

An appreciation of the correct choice of addendum modification on the slide to roll ratio, as reflected in the pitting resistance geometry factor, can be obtained by superimposing Figures 7.8 to 7.11 inclusive, as shown in Figure 7.12.

From Figure 7.12 it will be seen that when the gears are cut with no addendum modification, $S_P - S_G = 3.85$ and $I = 0.112$. However, when the addendum modifications are $x_P = -x_G = 0.372$, $S_P - S_G = 0.00$ and $I = 0.135$, representing an increased pitting resistance power rating of approximately 20%.

It has been the author's experience that when an existing gear set is exhibiting marked signs of wear, a change in the addendum modifications (to equalise the slide to roll ratio) has on occasions rectified the problem. Hence an increased pitting resistance power rating would appear to be justified. As to the order of magnitude of any increase, each reader will have their own opinion, each of which would be difficult to quantify.

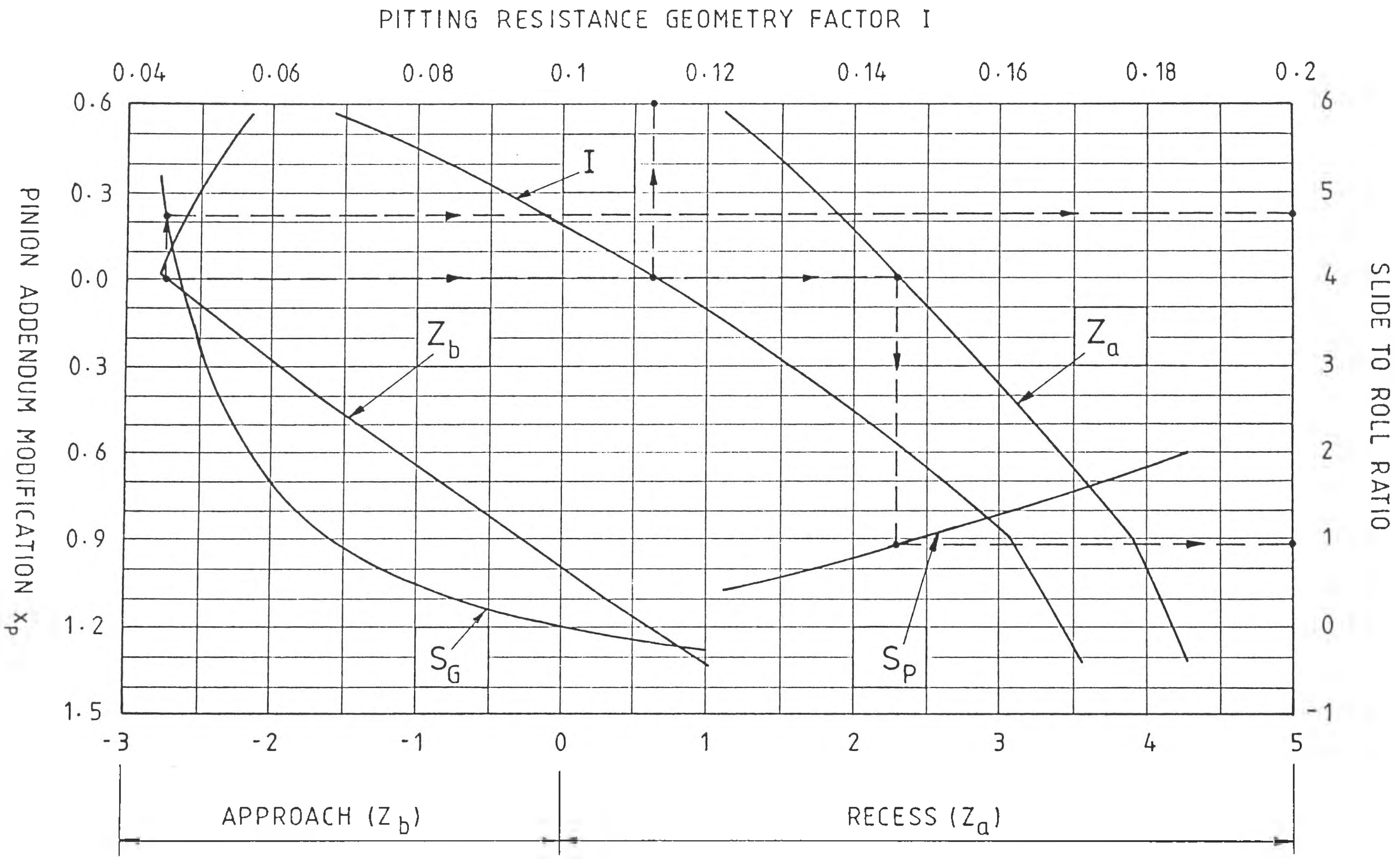


FIGURE 7·12 GEOMETRY FACTOR I AS A FUNCTION OF SLIDE TO ROLL RATIO

The problem is further exacerbated, if in the example being considered, the addendum modifications were $x_P = -x_G = +1.330$. In this instance, $I = 0.171$, representing an increase in the pitting resistance power rating of approximately 52%, whilst $S_P - S_G = 2.26$. It should be noted, that in order to meet the set constraints for C_I, C_U, C_W and m_T, the pinion and wheel were truncated 0.191 mm and 0.003 mm respectively.

This observation indicates that the geometry factor for pitting resistance, I, and consequently the wear rating, is independent of the slide to roll ratio.

To understand the relationship between I, and the slide to roll ratio, it would appear prudent to derive the AGMA 218.01 pitting resistance power rating equation from first principles.

From Figure 7.13 it can be seen that

$$P_{ac} = f(W_n, d, n_P)$$

$$= \frac{2\,\pi\, n_P\, d\, W_n \cos(\phi_t)}{60 \times 2 \times 10^3 \times 10^3}$$

$$= \frac{n_P\, d\, W_n \cos(\phi_t)}{1.91 \times 10^7} \qquad (7.15)$$

The classical treatment of the stress conditions and deformation at the surfaces of elastic bodies making point or line contact is due to Hertz. The formulae for the case of cylinders in line contact are as follows:

$$s_{ac} = C_p \left[\frac{W_n}{F} \left(\frac{1}{R_1} + \frac{1}{R_2} \right) \right]^{0.5} \qquad (7.16)$$

$$\text{where } C_p = [\pi\{(1-\mu_1^2)/E_1 + (1-\mu_2^2)/E_2\}]^{-0.5} \qquad (7.17)$$

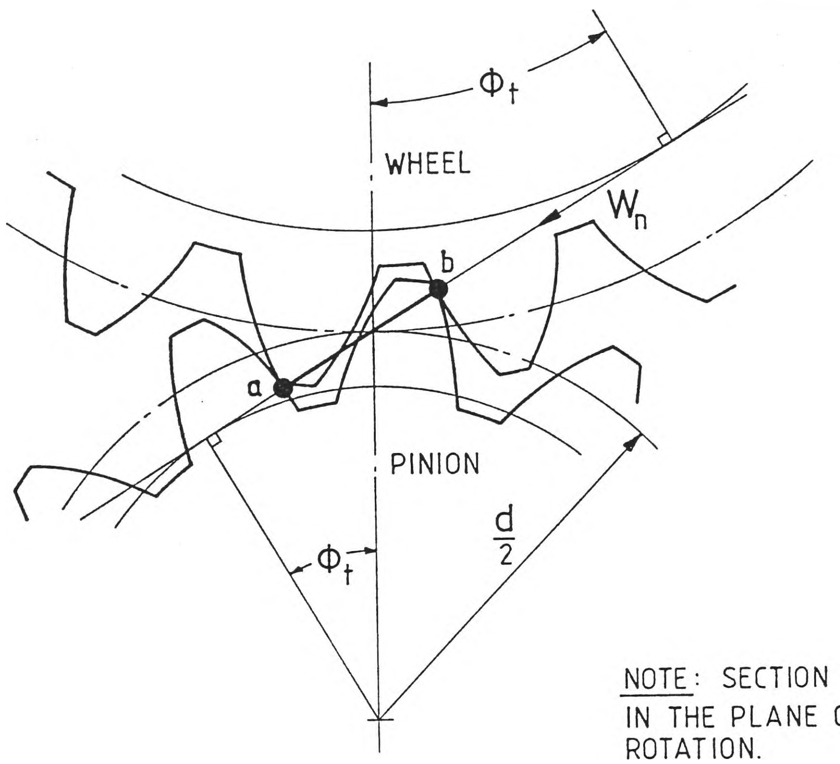


FIGURE 7·13 TOOTH CONTACT IN TRANSVERSE PLANE

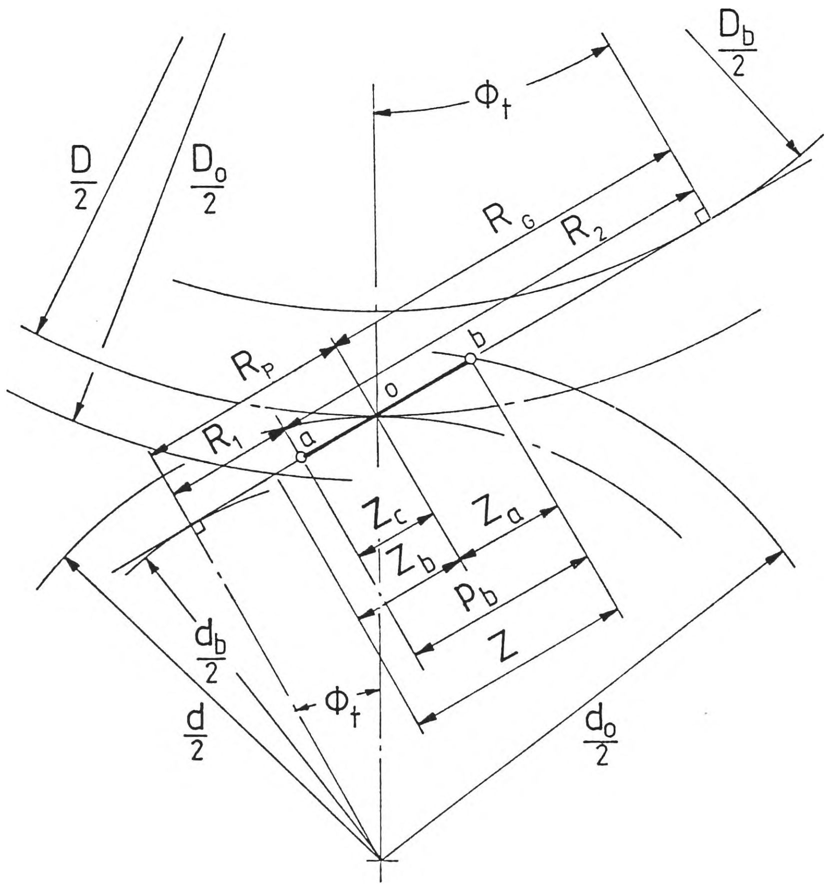


FIGURE 7·14 DIAGRAM FOR HERTZIAN RADII R_1 AND R_2

Rearranging equation (7.16) yields

$$W_n = \left[\frac{s_{ac}}{C_p} \right]^2 \left[\frac{R_1 R_2}{R_1 + R_2} \right] F \tag{7.18}$$

From Figure 7.14 it can be seen that

$$R_1 + R_2 = R_P + R_G \tag{7.19}$$

$$R_P = d \sin(\phi_t)/2 \tag{7.20}$$

$$R_G = D \sin(\phi_t)/2$$

But $D = m_G\, d$

$$\therefore \quad R_G = m_G\, d \sin(\phi_t)/2 \tag{7.21}$$

Combining equations (7.19) to (7.21) yields

$$R_1 + R_2 = d \sin(\phi_t)[m_G + 1]/2 \tag{7.22}$$

Combining equations (7.18) and (7.22) yields

$$W_n = \left[\frac{s_{ac}}{C_p} \right]^2 \frac{2\, F\, R_1\, R_2}{d \sin(\phi_t)[m_G + 1]} \tag{7.23}$$

Let $C_x = \dfrac{R_1\, R_2}{R_P\, R_G}$ (7.24)

Combining equations (7.20), (7.21) and (7.24) yields

$$C_x = \frac{4\, R_1\, R_2}{d^2\, m_G \sin^2(\phi_t)} \tag{7.25}$$

$$\text{Let } C_c = \frac{\cos(\phi_t)\sin(\phi_t)}{2} \quad \frac{m_G}{[m_G + 1]} \tag{7.26}$$

$$\text{and } I = C_c \; C_x \tag{7.27}$$

Combining equations (7.25) to (7.27) yields

$$I = \frac{2\, R_1\, R_2}{d^2 [m_G + 1]\tan(\phi_t)} \tag{7.28}$$

Combining equations (7.23) and (7.28) yields

$$W_n = \left[\frac{s_{ac}}{C_p} \right]^2 \frac{d\; F\; I}{\cos(\phi_t)} \tag{7.29}$$

Combining equations (7.15) and (7.29) yields

$$P_{ac} = \frac{n_P\; F\; I}{1.91 \times 10^7} \left[\frac{d\; s_{ac}}{C_p} \right]^2 \tag{7.30}$$

which is directly comparable with equation (5.5M) of AGMA 218.01, when the derating factors are included.

In considering the pitting resistance power rating, P_{ac}, particular attention should be directed towards the geometry factor, I, as defined by equation (7.28).

For a particular gear set m_G is a constant, whilst d and ϕ_t are approximately constant, only varying if the sum of the addendum modification coefficients is not zero. Hence one may conclude that the geometry factor for pitting resistance, I, is approximately proportional to the product of R_1 and R_2. From equation (7.19) it will be observed that the sum of R_1 and R_2 is also approximately constant, as R_P and R_G are a function of d and ϕ_t.

Hence mathematically, the technique to maximise I, is one of selecting two numbers such that their product is a maximum, whilst the sum of the two selected numbers must be constant. This can be achieved by selecting two equal numbers.

The conclusion to be drawn from this axiom, is to increase R_1, whilst reducing R_2, remembering that R_1 and R_2 are analogous to the radii of the two Hertzian cylinders from which the pitting resistance power rating was derived. Figure 7.11 illustrates this observation, in that an increase in the pinion addendum modification coefficient has the direct effect of increasing R_1, as the base pitch, p_b, is a constant.

With reference to Figure 7.14, it will be observed that if R_1 equals R_p, then the equivalent Hertzian cylinders will contact at the pitch point. For the particular spur gear example being considered, this phenomenon will occur when,

$$r_o = [\{r \sin(\phi_t) + p_b\}^2 + \{r \cos(\phi_s)\}^2]^{0.5}$$

$$= [\{10 \sin 20^o + \pi \cos 20^o\}^2 + \{10 \cos 20^o\}^2]^{0.5}$$

$$= 11.354$$

As the outside radius of the unmodified pinion is 11.0 mm, then an addendum modification coefficient of + 0.354, will achieve an increase in R_1, such that the point of mesh is at the pitch point.

Figure 7.12 indicates that as the addendum modification coefficient of the pinion increases from - 0.57 to + 0.354, the pitting resistance geometry factor increases and the absolute value of the difference in the slide to roll ratios decreases. However, as x_P is increased beyond + 0.372, I continues to increase, whilst $S_G - S_P$ increases. This appears to be an inconsistent piece of logic, in that one obtains a benefit in the wear rating as a function of a reduction in the slide to roll ratio up to the pitch point, but beyond this point, the benefit continues whilst the slide to roll ratio deteriorates.

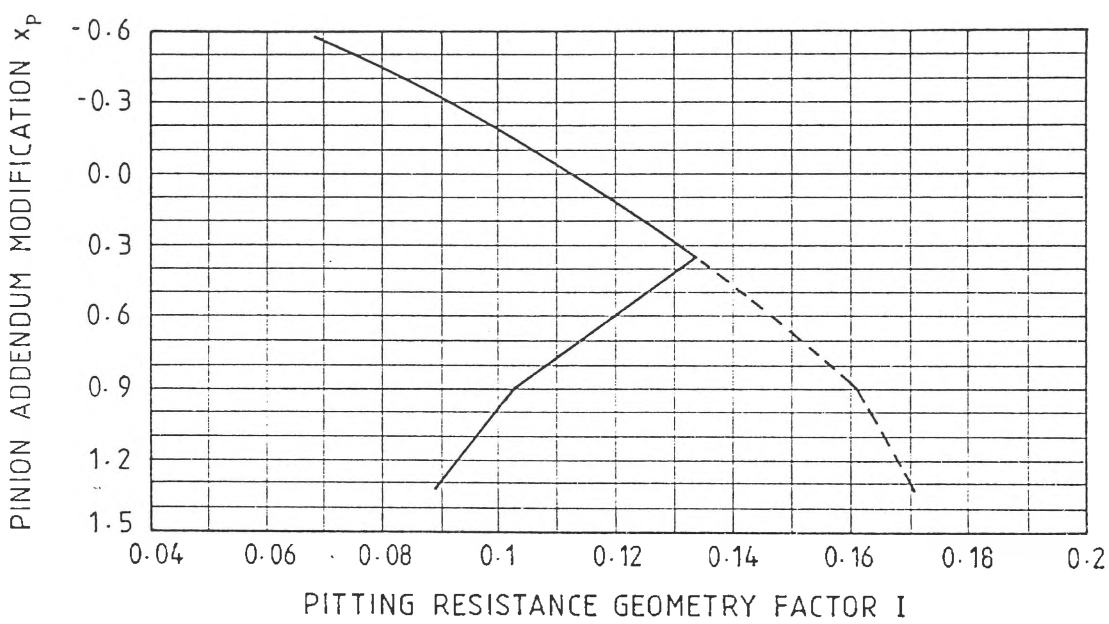


FIGURE 7.15 GEOMETRY FACTOR I AS A FUNCTION OF $|Z_C|$

It would appear to be better if the pitting resistance geometry factor decreased as R_1 was extended past the pitch point, thus bearing a symmetrical correlation to the slide to roll ratio. The AGMA 218.01 variable controlling this phenomenon is Z_c, the distance on the line of action from the pitch point to the point of stress calculation. If equation (6.14) of AGMA 218.01 were to be modified to incorporate the absolute value of Z_c, then a symmetrical correlation between the pitting resistance geometry factor and the slide to roll ratio could be achieved. The effect of this modification is shown in Figure 7.15.

For clarification, the two Hertzian cylinders have been added to Figure 7.14, together with the velocity diagrams, the combination being shown in Figure 7.16.

By an analysis similar to that of equations (7.1), (7.2) and (7.3), the specific sliding or slide to roll ratio, S_G, for the tip of the wheel, at point "a" on the pinion tooth is,

$$S_G = V_s/V_{r1}$$

$$S_G = \frac{R_2\ N_P}{R_1\ N_G} - 1 \tag{7.31}$$

It is apparent from Figure 7.16, that the slide to roll ratio will vary along the path of contact. Conventionally, the slide to roll ratio is calculated at the extremes of the path of contact. Hence, if in Figure 7.16, point "a" is at the end of the path of contact, such that ab = Z, then,

$$R_2 = [R_o^2 - R_b^2]^{0.5} \tag{7.32}$$

$$R_1 = C\sin(\phi_t) - R_2 \tag{7.33}$$

Combining equations (7.31), (7.32) and (7.33) yields

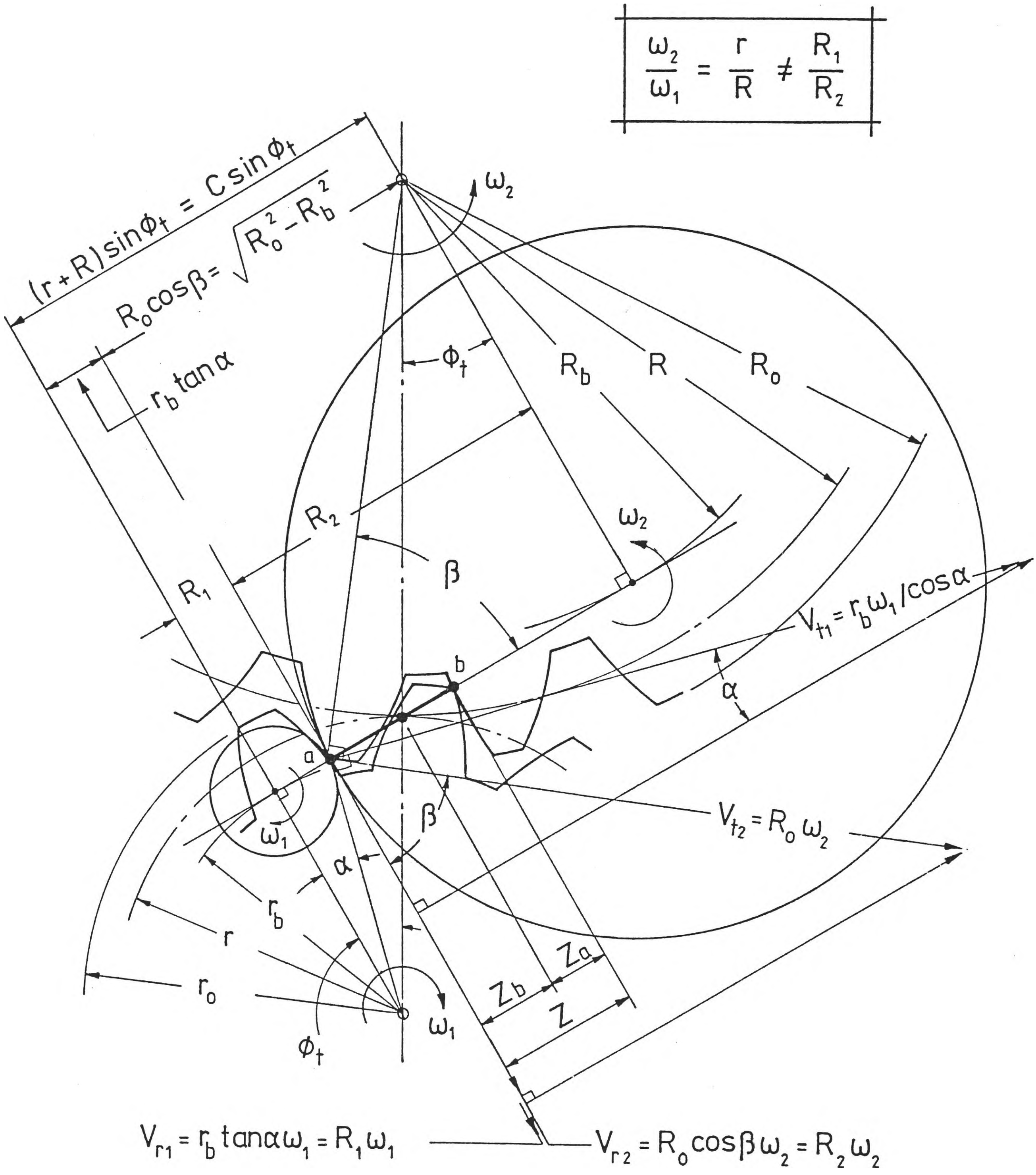


FIGURE 7.16 SPECIFIC SLIDING OR SLIDE TO ROLL RATIO

$$S_G = \frac{N_P[R_o^2 - R_b^2]^{0.5}}{N_G\{C \sin(\phi_t) - [R_o^2 - R_b^2]^{0.5}\}} - 1 \qquad (7.34)$$

Similarly, the slide to roll ratio S_P, for the tip of the pinion, when point "b" is at the end of the path of contact is

$$S_P = \frac{N_G[r_o^2 - r_b^2]^{0.5}}{N_P\{C \sin(\phi_t) - [r_o^2 - r_b^2]^{0.5}\}} - 1 \qquad (7.35)$$

Rearranging equation (7.34) with $m_G = N_G/N_P$ gives

$$S_G = \frac{(1 + m_G)[R_o^2 - R_b^2]^{0.5} - m_G\ C \sin(\phi_t)}{m_G\{C \sin(\phi_t) - [R_o^2 - R_b^2]^{0.5}\}}$$

$$S_G = \frac{(1 + m_G)\{[R_o^2 - R_b^2]^{0.5} - R \sin(\phi_t)\}}{m_G\{r \sin(\phi_t) + R \sin(\phi_t) - [R_o^2 - R_b^2]^{0.5}\}} \qquad (7.36)$$

From Figure 7.16 it can be shown that

$$Z_b = [R_o^2 - R_b^2]^{0.5} - R \sin(\phi_t) \qquad (7.37)$$

Combining equations (7.36) and (7.37) yields

$$S_G = \frac{[1 + m_G]\ Z_b}{R \sin(\phi_t) - m_G Z_b} \qquad (7.38)$$

which may be compared to equation (7.8) when $x = -Z_b$.

Similarly, equation (7.35) can be shown to be

$$S_P = \frac{[1 + m_G]\ Z_a}{R \sin(\phi_t) - Z_a} \qquad (7.39)$$

which may be compared to equation (7.9) when $x = Z_a$.

Hence it can be seen from equations (7.38) and (7.39) that the slide to roll ratio is a function of the approach and recess paths. Similarly equations (6.2), (6.7), (6.11), (6.12) and (6.14) of AGMA 218.01 show that the geometry factor for pitting resistance, I, is a function of the recess path, Z_a.

As the slide to roll ratio has an effect on the scoring and scuffing wear failure mode, one would expect to find a correlation between the slide to roll ratio, and the pitting resistance.

The suggested modification for spur gears, is to only consider the absolute value of the variable Z_c, in the calculation of the pitting resistance geometry factor, I.

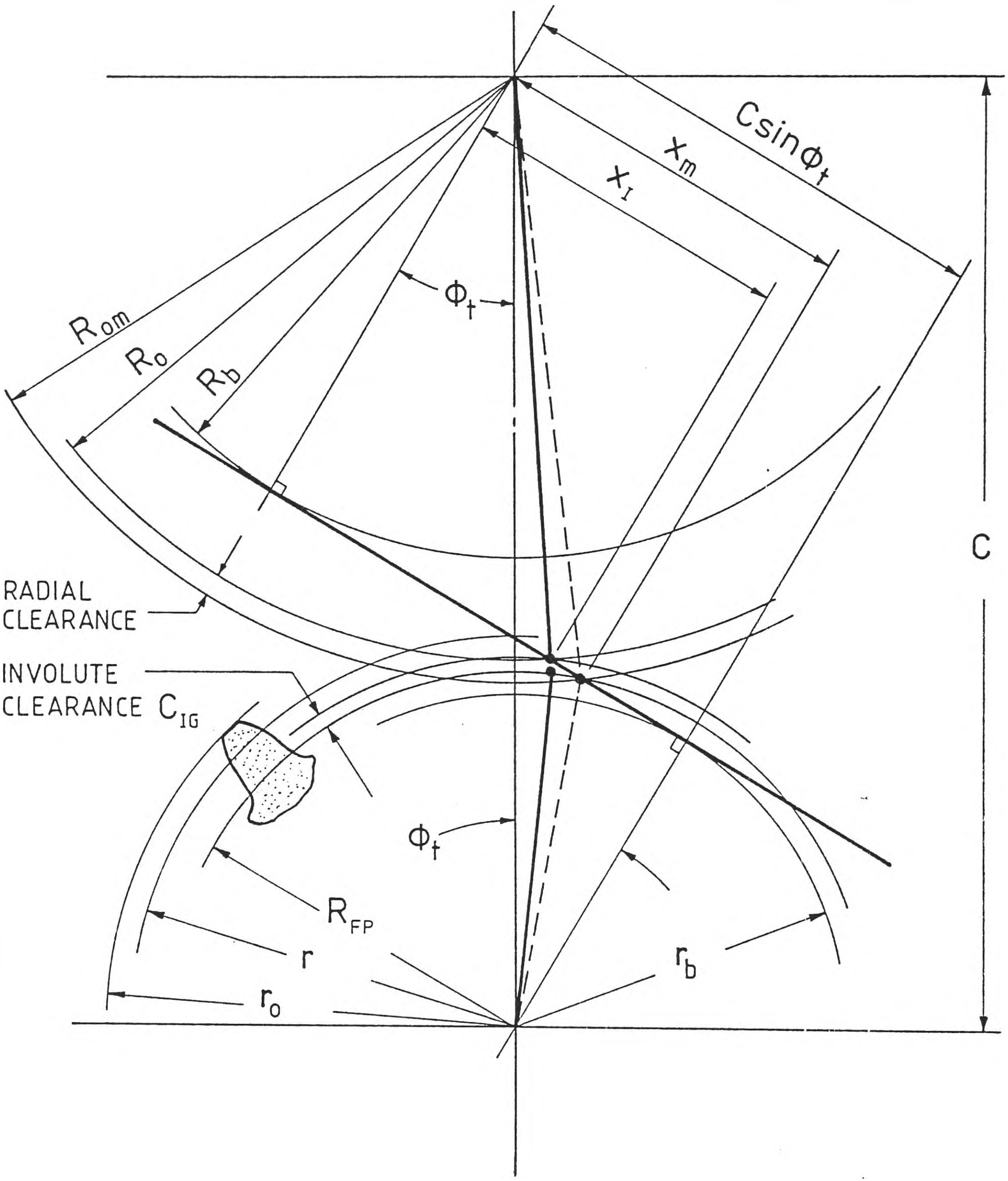


FIGURE 7.17 TIP TO ROOT FILLET INTERFERENCE

7.3 Tip to Root Fillet Interference

When addendum modifications are beyond those normally recommended, the potential exists for the tip of one gear and the root fillet of the other gear to come into contact. This situation is illustrated in Figure 7.17, from which one may calculate the maximum allowable tip radius of the wheel.

By inspection from Figure 7.17,

$$R_{FP} = [(C\ \sin(\phi_t) - x_m)^2 + r_b^2]^{0.5} \tag{7.40}$$

$$x_m = [R_{om}^2 - R_b^2]^{0.5} \tag{7.41}$$

Rearranging and solving for R_{om} yields

$$R_{om} = [\{C\ \sin(\phi_t) - (R_{FP}^2 - r_b^2)^{0.5}\}^2 + R_b^2]^{0.5} \tag{7.42}$$

Similarly,

$$r_{om} = [\{C\ \sin(\phi_t) - (R_{FG}^2 - R_b^2)^{0.5}\}^2 + r_b^2]^{0.5} \tag{7.43}$$

Having calculated the maximum allowable tip radius, the actual tip radius of the gear must be less. Experience has shown, that if the involute clearance per unit module is approximately 0.05, then the gear set will operate satisfactorily. An inspection of Figure 7.17 shows that the involute clearance is less than $R_{om} - R_o$.

The involute clearance is

$$C_{IG} = [(C\ \sin(\phi_t) - x_I)^2 + r_b^2]^{0.5} - R_{FP} \tag{7.44}$$

$$x_I = [R_o^2 - R_b^2]^{0.5} \tag{7.45}$$

Rearranging gives

$$C_{IG} = [\{C\ \sin(\phi_t) - (R_o^2 - R_b^2)^{0.5}\}^2 + r_b^2]^{0.5} - R_{FP} \tag{7.46}$$

Similarly,

$$C_{IP} = [\{C \sin(\phi_t) - (r_o^2 - r_b^2)^{0.5}\}^2 + R_b^2]^{0.5} - R_{FG} \qquad (7.47)$$

If C_{IP} or C_{IG} is less than 0.05 per unit module, then it is suggested that the outside radius of the offending gear be reduced.

This reduction in the outside radius is sometimes referred to as "topping", or more correctly, radial tooth truncation. In order to calculate the required amount of radial tooth truncation, the preceding equations must be rearranged, to obtain a specified C_I.

Recalling equations (7.44) and (7.45) and rearranging similar to equation (7.42) gives

$$C_{IG} = [(C \sin(\phi_t) - x_I)^2 + r_b^2]^{0.5} - R_{FP} \qquad (7.44)$$

$$x_I = [R_o^2 - R_b^2]^{0.5} \qquad (7.45)$$

$$R_o = [\{C \sin(\phi_t) - ((R_{FP} + C_{IG})^2 - r_b^2)^{0.5}\}^2 + R_b^2]^{0.5} \qquad (7.48)$$

Similarly,

$$r_o = [\{C \sin(\phi_t) - ((R_{FG} + C_{IP})^2 - R_b^2)^{0.5}\}^2 + r_b^2]^{0.5} \qquad (7.49)$$

It is of interest to note, that whilst Buckingham (9) has developed similar equations, he has a different set of equations, which are unique for a rack and pinion gearset. However, when the computer program of APPENDIX A has 100,000 "wheel teeth" to simulate a rack, equations (7.46) and (7.47), yield involute clearances commensurate with those from Buckingham's equations for rack and pinion gear sets.

A computer analysis utilising the preceding equations may be found in Section 7.1 of APPENDIX C.

7.4 Top Land Width Coefficient

The top land width coefficient, C_W, controls the width of the tooth at the tip to ensure that there is sufficient thickness. For gears that are to be surface heat treated, the tip thickness should be increased in conformity with good engineering practice. Experience has shown, that if the top land width coefficient per unit module is approximately 0.4, then the gear design will be satisfactory for most applications.

The width of the crest or top land of a tooth, measured as a chord in the normal plane, may be calculated as follows

1. Calculate the transverse reference arc tooth thickness t_{st}.

$$t_{st} = \{m_n[0.5\pi + 2x \tan(\phi_c)] - B_N/\cos(\phi_c)\}/\cos(\psi_s) \qquad (7.50)$$

2. Calculate the tip helix angle ψ_o.

$$\psi_o = \tan^{-1}[r_o \tan(\psi_s)/r_s] \qquad (7.51)$$

where r_s is the reference pitch radius.

3. Calculate the tip transverse pressure angle ϕ_o.

$$\phi_o = \cos^{-1}(r_b/r_o) \qquad (7.52)$$

4. Calculate the transverse arc tooth thickness at the top land t_{ot}.

$$t_{ot} = 2r_o[0.5\ t_{st}/r_s + \mathrm{inv}(\phi_s) - \mathrm{inv}(\phi_o)] \qquad (7.53)$$

5. Calculate the projected transverse half tip angle Ω_h.

$$\Omega_h = 0.5\ t_{ot} \cos^2(\psi_o)/r_o \qquad (7.54)$$

6. Calculate the top land width in the normal plane t_o.

$$t_o = 2r_o \sin(\Omega_h)/\cos(\psi_o) \qquad (7.55)$$

7. Calculate the top land width coefficient C_W.

$$C_W = t_o/m_n \tag{7.56}$$

However, if it is desired to find the value of r_o to give a required crest width, then equations (7.51) to (7.55) inclusive must be solved by a trial and error process. This procedure may be accelerated by the use of Newton's Method for iteration.

Combining equations (7.52) and (7.53) gives

$$\frac{t_{ot}}{2r_b} = \frac{0.5\ t_{st}}{r_s \cos(\phi_o)} + \frac{\mathrm{inv}(\phi_s)}{\cos(\phi_o)} - \frac{\tan(\phi_o)}{\cos(\phi_o)} + \frac{\phi_o}{\cos(\phi_o)} \tag{7.57}$$

From equations (7.55), (7.56) and (7.57)

$$\text{Let } C_1 = 0.5\ m_n\ C_W/[r_b \cos(\Psi_o)] \tag{7.58}$$

$$\text{and } C_2 = 0.5\ t_{st}/r_s + \mathrm{inv}(\phi_s) \tag{7.59}$$

$$f(\phi_o) = C_2 \sec(\phi_o) - \tan(\phi_o) \sec(\phi_o) + \phi_o \sec(\phi_o) - C_1 \tag{7.60}$$

Now $\quad f'[\sec(\phi_o)] = \tan(\phi_o) \sec(\phi_o)$

and $\quad f'[\tan(\phi_o)] = \sec^2(\phi_o)$

$$\therefore \quad f'(\phi_o) = C_2 \tan(\phi_o) \sec(\phi_o) - [\tan^2(\phi_o) \sec(\phi_o) + \sec^3(\phi_o)]$$

$$+ [\phi_o \tan(\phi_o) \sec(\phi_o) + \sec(\phi_o)]$$

$$= \sec(\phi_o)[C_2 \tan(\phi_o) - \tan^2(\phi_o) - \sec^2(\phi_o) + \phi_o \tan(\phi_o) + 1]$$

Now $\quad 1 - \sec^2(\phi_o) = -\tan^2(\phi_o)$

$$\therefore \quad f'(\phi_o) = \sec(\phi_o) \tan(\phi_o)[C_2 - 2 \tan(\phi_o) + \phi_o] \tag{7.61}$$

Newton's Method of iteration may be expressed as

$$\phi_{o[n+1]} = \phi_{o[n]} - f(\phi_{o[n]})/f'(\phi_{o[n]}) \tag{7.62}$$

Combining equations (7.60), (7.61) and (7.62) yields

$$\phi_{o[n+1]} = \phi_{o[n]} - \frac{C_2 - \tan(\phi_{o[n]}) - \phi_{o[n]} - C_1 \cos(\phi_{o[n]})}{\tan(\phi_{o[n]})[C_2 - 2\tan(\phi_{o[n]}) + \phi_{o[n]}]} \tag{7.63}$$

The procedure to find r_o for a specified C_W is as follows

1. For the existing gear calculate t_{st} from equation (7.50).

2. For the existing gear calculate r_o and r_s and then calculate ψ_o and ϕ_o from equations (7.51) and (7.52) respectively.

3. Calculate C_1 and C_2 from equations (7.58) and (7.59) respectively.

4. Iterate equation (7.63) where $\phi_{o[1]}$ is the value of ϕ_o calculated in Step 2.

5. Calculate the new r_o from equation (7.52). For helical pinions, it may be necessary to repeat steps 2 to 5, with the new value of r_o substituted for the original value of r_o.

6. Calculate the new C_W from equations (7.51) to (7.56) inclusive.

7.5 An Example of the Use of the Computer Program

WOLLONGONG UNIVERSITY - SCHOOL OF MECHANICAL ENGINEERING
ANALYST: ROBERT DAVEY DATE: 9TH JAN 1989
THESIS EXAMPLE - HELICAL GEARS

SPECIFIED RANGES OF DESIGN REQUIREMENTS.

STEP	SYMBOL	MINIMUM	MAXIMUM	UNITS	DESCRIPTION
I	PACP	100.000	120.000	KW	WEAR POWER RATING PINION
N	PACW	100.000	120.000	KW	WEAR POWER RATING WHEEL
P	PATP	100.000	120.000	KW	STRENGTH POWER RATING PINION
U	PATW	100.000	120.000	KW	STRENGTH POWER RATING WHEEL
T	C	150.000	160.000	MM	CENT DISTANCE STEPS OF 1.00 MM
	F	50.000	60.000	MM	FACE WIDTH STEPS OF 1.00 MM
D	RPMP	1440.000	1440.000	RPM	REVOLUTIONS / MINUTE PINION
A	NP	25	30		NUM. PINION TEETH STEPS OF 1
T	MG	2.600	2.700		GEAR RATIO
A	PSIS	10 00'00"	15 00'00"	DMS	HELIX ANGLE STEPS OF 1.0 DEG
	LIFEP	40000.000	40000.000	HOURS	DESIGN LIFE PINION
	LIFEW	45000.000	45000.000	HOURS	DESIGN LIFE WHEEL
	SHP	450.000	60.000	BHN/RC	SURFACE HARDNESS OF PINION
	SHW	300.000	55.000	BHN/RC	SURFACE HARDNESS OF WHEEL

SPECIFIED DESIGN FACTORS OF AGMA 218.01 (DEC 1982).

STEP	SYMBOL	PINION	WHEEL	CLAUSE	DESCRIPTION
I	QV	8	6	8.3	QUALITY NUMBER FROM AGMA 390
N	CA	1.000	1.000	9.1	APPLIC. FCT. FOR PITTING RES.
P	KA	1.000	1.000	9.1	APPLIC. FCT. FOR BENDING STR.
U	CF	1.000	1.000	11.0	SURFACE CONDITION FACTOR
T	CS	1.000	1.000	12.2	SIZE FCT. FOR PITTING RES.
	KS	1.000	1.000	12.2	SIZE FCT. FOR BENDING STR.
D	CMT	1.000	1.000	13.2.1	TRANSVERSE LOAD DISTRIB. FCT.
A	CMC	1.000	1.000	13.2.2	LEAD CORRECTION FACTOR
T	CPM	1.100	1.100	13.2.2.1	PINION PROPORTIONAL MODIFIER
A	CE	1.000	1.000	13.2.2.1	MESH ALIGNMENT CORRECTION FCT.
	FP	64.000	64.000	15.2	SURFACE FINISH (MICROINCHES)
	CR	1.000	1.000	17.0	RELIABILITY FCT. PITTING RES.
	KR	1.000	1.000	17.0	RELIABILITY FCT. BENDING STR.
	CT	1.000	1.000	18.0	TEMPERATURE FCT. PITTING RES.
	KT	1.000	1.000	18.0	TEMPERATURS FCT. BENDING STR.
	IDLER	2	1	14.4.1	NUMBER OF GEARS IN CONTACT
	BUTT	YES	NO	6.3.2.3	BUTTRESSING OF GEAR
	ROUGH	NO	NO	TABLE 2	INACCURATE SPUR GEARS

SPECIFIED LIMITS FOR ADDENDUM MODIFICATION COEFFICIENTS.

		1	2	3	4
CURVE	2	OPEN GEARING	COMMERCIAL ENCLOSED	PRECISION ENCLOSED	EXTRA PRECISION
TOTAL	4	ISO RECOMMENDED	ISO CONVENTIONAL	B.S. RECOMMENDED	UNLIMITED
ALLOT	2	ISO DISTRIBUTION	B.S. DISTRIBUTION	EQUAL BND STRENGTH	EQUAL SLIDE/ROLL

CURVE = CURVE FOR ALIGNMENT FACTOR (AGME 218.01 FIG 10)
TOTAL = SUMMATION OF X(1) & X(2)
ALLOT = DISTRIBUTION OF X(1) & X(2)
ISO = INTERNATIONAL STANDARD ISO/TR 4467 - 1982 (E).
B.S. = BRITISH STANDARD PD 6457 - NOV 1970.

WOLLONGONG UNIVERSITY - SCHOOL OF MECHANICAL ENGINEERING
ANALYST: ROBERT DAVEY DATE: 9TH JAN 1989
THESIS EXAMPLE - HELICAL GEARS
GEAR RATING TO AGMA 218.01 (DEC. 1982).

SYMBOL	DESCRIPTION	PINION	WHEEL
N	NUMBER OF TEETH	27	72
RPM	SPEED OF ROTATION (REV/MIN)	1440.000	540.000
F	FACE WIDTH (MM)	51.000	51.000
PSIS	HELIX ANGLE (DMS)	12 00'00"	12 00'00"
MN	NORMAL MODULE (MM)	3.000	3.000
MT	TRANSVERSE MODULE (MM) MT=MN/COS(PSIS)	3.067	3.067
D	OPERATING PITCH CIRCLE DIAMETER (MM)	84.545	225.455
K10	F * RPMP * DP / (1.91*10**7)	.325	.325
IDLER	NUMBER OF GEARS IN CONTACT	2	1
I	GEOMETRY FACTOR FOR PITTING RESISTANCE	.210	.210
J	GEOMETRY FACTOR FOR BENDING STRENGTH	.711	.625
SH	SURFACE HARDNESS (RC)	55.000	50.000
VT	PI * RPMP * DP / 60000 (M/SEC)	6.375	6.375
QV	QUALITY NUMBER - AGMA 390	8	6
CV	FIG 7	.773	.679
KV	FIG 7	.773	.679
CA	CLAUSE 9.1	1.000	1.000
KA	CLAUSE 9.1	1.000	1.000
CP	TABLE 4 (MPA ** 0.5)	189.809	189.809
CF	CLAUSE 11.1	1.000	1.000
CS	CLAUSE 12.2	1.000	1.000
KS	CLAUSE 12.2	1.000	1.000
CMT	CLAUSE 13.2.1	1.000	1.000
CMC	CLAUSE 13.2.2.1	1.000	1.000
F/D	FACE WIDTH TO DIAMETER RATIO	.603	.226
CPF	FIG 8	.048	.048
CPM	CLAUSE 13.2.2.1	1.100	1.100
CMA	FIG 10 (CURVE 2)	.158	.158
CE	CLAUSE 13.2.2.1	1.000	1.000
CMF	1.0 + CMC * (CPF*CPM+CMA*CE)	1.211	1.211
CM	CMF * CMT	1.211	1.211
KM	CMF * CMT	1.211	1.211
SAC	TABLE 5 (MPA)	1250.000	1200.000
SAT	TABLE 6 (MPA)	380.000	310.000
CH	FIG 18	1.000	1.000
N	NUMBER OF LOAD CYCLES * 10**6	6912.000	1458.000
CL	FIG 20	.805	.892
KL	FIG 21	.874	.931
CR	TABLE 8	1.000	1.000
KR	TABLE 8	1.000	1.000
CT	CLAUSE 18.1	1.000	1.000
KT	CLAUSE 18.1	1.000	1.000
PITTING PAC (KW)	$\frac{K10 \cdot I \cdot CV \cdot DP}{CS \cdot CM \cdot CF \cdot CA} \left(\frac{SAC \cdot CL \cdot CH}{CP \cdot CT \cdot CR} \right)^2$	103.418	102.816
BENDING PAT (KW)	$\frac{K10 \cdot J \cdot KV \cdot SAT \cdot KL \cdot MT}{KA \cdot KS \cdot KM \cdot KR \cdot KT}$	105.189	100.814

SECTION 8

EXPERIMENTAL RESULTS

8.1 Introduction

As part of the ongoing programme of gear research being conducted at the University of Wollongong, a gear testing rig has been manufactured. The major aims of the tests carried out on the gear testing rig are to correlate both spur and helical gear power ratings to various national codes and to examine the effect of modified addenda on the performance of gears.

A comparison of gear rating procedures is particularly relevant at present, since the original Australian Standard B61 (22) (an endorsement without modification of BS 436 (21)) has been replaced by AS 2938(5). The current Australian Standard, AS 2938, is based on AGMA 218.01(4). This Section will compare AS B61 power ratings of the four test gear sets with the power ratings as given by AGMA 218.01. Further, the introduction of the draft ISO International Standards 6336/1(23) and 6336/2(24) for the calculation of the surface durability has further compounded the problem.

To examine the effect of modified addenda, the gear testing rig includes two spur gear sets, identical in all respects, with the exception that one set is cut with zero addendum modifications, whilst the other set has addendum modifications to Clause 43 of BS 436. A conventional helical gear set and a third spur gear set designed for limited life complete the testing rig.

Because of energy considerations, and the fact that the gear testing rig was designed and commissioned in 1979 and 1980, when the Australian Standard for gear rating was AS B61, each gear set was designed to have a BS 436 nominal power rating of 2 kW.

FIGURE 8.1 - THE GEAR TESTING RIG

8.2 The Gear Testing Rig

The gear testing rig is shown in Figure 8.1.

The gear testing rig comprises four sets of 2 kW 415 V AC motors each of which transmits power through a gear pair to a DC motor connected up as a generator. The power is dissipated in a resistor bank which provides the load to the system.

Coupling between driving and driven components is by means of coneflex rubber couplings. Braking of gear sets 1-3 is achieved by means of electromagnetic brakes, whilst gear set 4 is braked by dynamic braking.

An integral electrical control system is utilised to maintain equilibrium of the system and to prevent excessive power loadings during start-up. The control system also has the ability to reverse direction of the motors on a cyclic basis as well as to monitor time in service of each gear set and provide overload protection in the event of system failures occurring.

A BS 436 Class A2 cutter was utilised in the manufacture of all four gear sets, details of which are shown in Table 8.1. The symbols shown in Table 8.1 are AGMA 218.01 notation.

Description	Gear Set 1		Gear Set 2		Gear Set 3		Gear Set 4	
	Pinion	Wheel	Pinion	Wheel	Pinion	Wheel	Pinion	Wheel
Number of teeth, N_P and N_G	71	355	71	355	29	145	24	119
Gear Ratio, m_G	5.00		5.00		5.00		4.96	
Rotational speed, n_P and n_G (rpm)	1440	288	1440	288	1440	288	1440	290
Helix angle, ψ (deg)	0.0	0.0	0.0	0.0	37.0	37.0	0.0	0.0
Normal module, m_n (mm)	1.000	1.000	1.000	1.000	1.000	1.000	1.500	1.500
Operating P.C.D., d and D (mm)	71.000	355.000	71.000	355.000	36.312	181.560	36.000	178.500
Operating centre distance, C (mm)	213.000		213.000		108.936		107.250	
Facewidth, F (mm)	11.0	11.0	11.0	11.0	25.0	25.0	12.0	12.0
Addendum modification (mm)	0.320	-0.320	0.0	0.0	0.320	-0.320	0.479	-0.479
Normal backlash (mm)	0.077	0.134	0.077	0.134	0.058	0.082	0.076	0.104
Constant chord height (mm)	1.030	0.465	0.748	0.748	1.030	0.465	1.544	0.698
Constant chord width (mm)	1.516	1.048	1.310	1.253	1.534	1.100	2.312	1.668
Material specification (25)	S1040	S1040	S1040	S1040	S1040	S1040	S1040	S1040
Tensile strength (MPa)	618	540	618	540	618	540	618	540
Brinell hardness (26)	179	152	179	152	179	152	179	152
Running time (hrs/day)	24	24	24	24	24	24	0.5	0.5

TABLE 8.1 - DETAILS OF THE GEAR SETS

It should be noted that of the four gear sets, only gear set 4 is a "hunting" ratio.

The choice of the small module for each gear set is as a result of the "optimum" design of the gear sets. The design procedure used was developed by Davey (27) and refined by Buchhorn (28) and produces a design which generates the minimum excess power capacity.

The gears themselves may be categorised as "commercial gears". They, and other components such as brakes and couplings, have been selected to simulate an industrial environment rather than a high precision, rigorously controlled test environment. The pinion and wheel materials are identical in chemical composition, whilst the pinion is slightly harder than the wheel. The gear material is a 40 carbon steel and corresponds to AS 1448/Grade S1040.

Lubrication of the gears is achieved using a splash lubrication system which has Mobil Grade 632 gear oil as the lubricant. The gear testing rig has two separate oil sumps; the first is common to gear sets 1, 2 and 3, whilst gear set 4 has an individual oil sump. This arrangement aims at removing the influence of lubrication when comparing the performances of gear sets 1, 2, and 3. Gear set 4 was expected to rapidly deteriorate and thus the foreign particles generated were separated from the other gear sets.

8.3 Power Ratings

The four gear sets have been rated to BS 436, AGMA 218.01, ISO 6336/1 and 6336/2, and the results are summarised in Tables 8.2 to 8.4. The symbols shown in Table 8.2 are BS 436 notation, whilst those in Tables 8.3 and 8.4 are AGMA 218.01 notation and ISO 6336 notation respectively.

The BS 436 strength rating is given by

$$P_b = \frac{X_b\ S_b\ Y\ F\ n_p\ N_p\ m_n^2}{1.91 \times 10^7\ \cos^2 \psi} \tag{8.1}$$

and the BS 436 wear rating is given by

$$P_c = \frac{X_c\ S_c\ Z\ F\ n_p\ N_p\ m_n^{1.8}}{10^7\ \cos^{1.8} \psi} \tag{8.2}$$

The power ratings of the four gear sets as predicted by BS 436 are given in Table 8.2.

Description		Gear Set 1 Pinion	Gear Set 1 Wheel	Gear Set 2 Pinion	Gear Set 2 Wheel	Gear Set 3 Pinion	Gear Set 3 Wheel	Gear Set 4 Pinion	Gear Set 4 Wheel
Bending stress factor, S_b	(MPa)	169	131	169	131	169	131	169	131
Surface stress factor, S_c	(MPa)	13.8	9.6	13.8	9.6	13.8	9.6	13.8	9.6
Strength speed factor, X_b		0.225	0.31	0.225	0.31	0.225	0.31	0.39	0.55
Wear speed factor, X_c		0.195	0.266	0.195	0.266	0.195	0.266	0.70	0.98
Strength factor, Y		0.91	0.82	0.86	0.88	0.68	0.59	0.76	0.66
Zone factor, Z		6.14	6.14	5.90	5.90	4.82	4.82	2.58	2.58
Strength rating, P_b	(kW)	2.04	1.96	1.93	2.10	2.22	2.05	2.45	2.32
Wear rating, P_c	(kW)	1.86	1.76	1.79	1.69	2.03	1.93	2.14	2.09

TABLE 8.2 - BS 436 POWER RATINGS

The AGMA 218.01 strength rating is given by

$$P_{at} = \frac{n_P\, d\, K_v}{1.91 \times 10^7\, K_a} \cdot \frac{F\, m_n}{\cos \psi} \cdot \frac{J}{K_s K_m} \cdot \frac{s_{at} K_L}{K_R\, K_T} \tag{8.3}$$

and the AGMA 218.01 pitting resistance rating is given by

$$P_{ac} = \frac{n_P\, F}{1.91 \times 10^7} \cdot \frac{I\, C_v}{C_s C_m C_f C_a} \cdot \left[\frac{d\, s_{ac} C_L C_H}{C_p C_T C_R}\right]^2 \tag{8.4}$$

The power ratings of the four gear sets as predicted by AGMA 218.01 are given in Table 8.3.

Description		Gear Set 1		Gear Set 2		Gear Set 3		Gear Set 4	
		Pinion	Wheel	Pinion	Wheel	Pinion	Wheel	Pinion	Wheel
Allowable bending stress number, s_{at}	(MPa)	180	170	180	170	180	170	180	170
Allowable contact stress number, s_{ac}	(MPa)	600	590	600	590	600	590	600	590
Elastic coefficient, C_p	$(MPa)^{1/2}$	187	187	187	187	187	187	187	187
Transmission accuracy level number, Q_v		8	8	8	8	8	8	8	8
Dynamic factors, C_v and K_v		0.79	0.79	0.79	0.79	0.84	0.84	0.84	0.84
Strength life factor, K_L		0.91	0.94	0.91	0.94	0.91	0.94	0.98	1.00
Pitting resistance life factor, C_L		0.87	0.90	0.87	0.90	0.87	0.90	0.95	0.99
Load distribution factors, C_m and K_m		1.16	1.16	1.16	1.16	1.19	1.19	1.16	1.16
Strength geometry factor, J		0.511	0.441	0.478	0.467	0.456	0.433	0.459	0.396
Pitting resistance geometry factor, I		0.138	0.138	0.131	0.131	0.245	0.245	0.134	0.134
Strength rating, P_{at}	(kW)	3.36	2.83	3.14	2.99	4.52	4.19	2.86	2.38
Pitting resistance rating, P_{ac}	(kW)	3.06	3.17	2.91	3.01	3.35	3.47	1.06	1.11

NOTE: The application factors C_a and K_a, reliability factors C_R and K_R, size factors C_s and K_s, temperature factors C_T and K_T, surface condition factor C_f and hardness ratio factor C_H are all taken to be unity for both the pinion and wheel in each of the four gear sets.

TABLE 8.3 - AGMA 218.01 POWER RATINGS

The ISO 6336/1 and 6336/2 surface durability (pitting) rating is given by

$$P_\sigma = \frac{m_G\ F\ d^2\ n_P}{1.91 \times 10^7 (m_G+1) K_A K_v K_{H\alpha} K_{H\beta}} \left[\frac{\sigma_{Hlim} Z_N Z_L Z_R Z_v Z_W Z_X}{S_{Hmin} Z_H Z_E Z_\varepsilon Z_\beta} \right]^2$$

The surface durability ratings of the four gear sets as predicted by ISO 6336/1 and 6336/2 are given in Table 8.4.

Description	Gear Set 1		Gear Set 2		Gear Set 3		Gear Set 4	
	Pinion	Wheel	Pinion	Wheel	Pinion	Wheel	Pinion	Wheel
Endurance limit, σ_{Hlim} (MPa)	450	400	450	400	450	400	450	400
Elasticity factor, Z_E (MPa)	190	190	190	190	190	190	190	190
ISO accuracy grade	7	7	7	7	7	7	7	7
Dynamic factor, K_v	1.35	1.35	1.35	1.35	1.35	1.04	1.05	1.05
Transverse load factor, $K_{H\alpha}$	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
Longitudinal load factor, $K_{H\beta}$	1.18	1.18	1.18	1.18	1.27	1.27	1.22	1.22
Zone factor, Z_H	2.49	2.49	2.49	2.49	2.09	2.09	2.49	2.49
Helix factor, Z_β	1.0	1.0	1.0	1.0	0.89	0.89	1.0	1.0
Contact factor, Z_ε	0.85	0.85	0.84	0.84	0.90	0.90	0.88	0.88
$Z_N Z_L Z_R Z_v Z_W Z_X$	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.2
Surface durability rating, P_σ (kW)	3.32	2.62	3.38	2.67	3.91	3.09	1.11	1.02

NOTE: 1. The application factor K_A, the safety factor for contact stress S_{Hmin} and the helix angle factor for contact stress, Z_β are all taken to be unity for both the pinion and wheel in each of the four gear sets.

2. $Z_N Z_L Z_R Z_v Z_W Z_X$ is the product of the life factor for contact stress, the lubricant factor, the roughness factor for contact stress, the speed factor, the work hardening factor and the size factor for contact stress respectively.

TABLE 8.4 - ISO 6336 SURFACE DURABILITY POWER RATINGS

8.4 Differences Between AGMA 218.01 and ISO 6336/DIN 3990

The following extract from the FZG- Report 1367 (29) prepared by the Technical University of Munich for the ISO Gear Rating Committee highlights the major theoretical differences between AGMA 218.01 and ISO 6336/DIN 3990 (30).

"In comparing the AGMA and DIN Standards it is important that differences in the calculation theory be identified. It is noted that the AGMA Standard pertains to pitting resistance and bending strength but does not address scuffing resistance. Also AGMA 218.01 does not apply to internal gears for the calculation of bending strength.

The basic difference between AGMA and ISO/DIN is the calculation of the geometry factor (J in AGMA, and Y_F, Y_S and Y_β in ISO/DIN). The following differences are noted:

i. AGMA includes the tensile stress from superposition of bending and compression. ISO/DIN accounts for just the bending in the normal stress calculation.

ii. AGMA utilises the Lewis parabola of constant stress for determination of the critical root section. It is noted that this method is dependent of load direction. ISO/DIN takes the distance between the points at which the tangents at an angle of 30 degrees to the centre line of the gear teeth contact the root fillets. The critical tooth thickness by this method is independent of load direction.

iii. AGMA considers two load application points. For accurate gears with good load sharing, the load is applied at the "highest point of single tooth contact". For gears considered to not have load sharing, the load is applied at the tip. ISO/DIN considers two cases if the load is applied at the tip and converted to an application at the "highest point of single pair contact" with a "contact ratio factor" (Y_ε) or the load is directly applied at the "highest point of single pair contact".

iv. The stress correction factor accounts for the "notch effect" of the root fillet. AGMA calculates the root fillet radius and determines the stress correction factor from photoelastic experiments done by Dolan and Broghamer (12). The ISO/DIN Standards calculate the fillet radius at the critical section and determine the stress correction factor (Y_S) by comparison with actual stresses (eg measured in miniature strain gauge experiments with steel gears, calculated with finite element method, integral equations).

The ratings for pitting resistance in AGMA and ISO/DIN are based on the formulae developed by Hertz for contact pressure between two curved surfaces. There is no basic theory difference."

For a more detailed discussion on the apparent absence of an analysis of scuffing in AGMA as opposed to ISO/DIN standards, reference should be made to Section 7.2.

FIGURE 8.2 - THE NIKON V-12 PROFILE PROJECTOR

8.5 Measurement of Gear Wear Using Photomicrographs

Because of the small physical size of the gear teeth in each gear set, it was not possible to assess wear by direct physical measurement such as calipers or vernier gauges. After investigating various procedures including the use of surface roughness meters and optical microscopy, the measurement procedure adopted was the use of plastic replicas of the gear teeth and a profile projector. The procedure involves taking a replica of the gear tooth in question and using a profile projector to produce a magnified image. Verniers on the profile projector are used to measure the constant chord thickness (and hence wear) of the tooth.

On each of the eight gears, two teeth located diametrically opposite each other are marked and numbered position 1 and position 2. At operating intervals of approximately 500 hours running time, a replica is prepared of the designated gear teeth to determine the reduction in constant chord thickness.

Appreciating that it is difficult to use single values of tooth thickness to characterise the wear of an entire tooth surface, the constant chord thickness was selected as the datum as suggested by Clause 44 of BS 436.

The actual measurement of the constant chord thickness is carried out on a Nikon Model V-12 Profile Projector as shown in Figure 8.2. The micrometer heads which control the cross hairs can be read to 0.001mm, once the image has been correctly focussed.

Tables 8.3 and 8.4 predict a power rating in terms of the pitting resistance which is different from the mode of wear being measured by the above technique. However, as it is difficult to quantify the measurement of pitting, the author chose to measure reduction in tooth thickness, which can be quantified, and then compare any apparent correlation. Fortunately, to date, the results have indicated a reasonable correlation between a reduction in tooth thickness and the propagation of pitting.

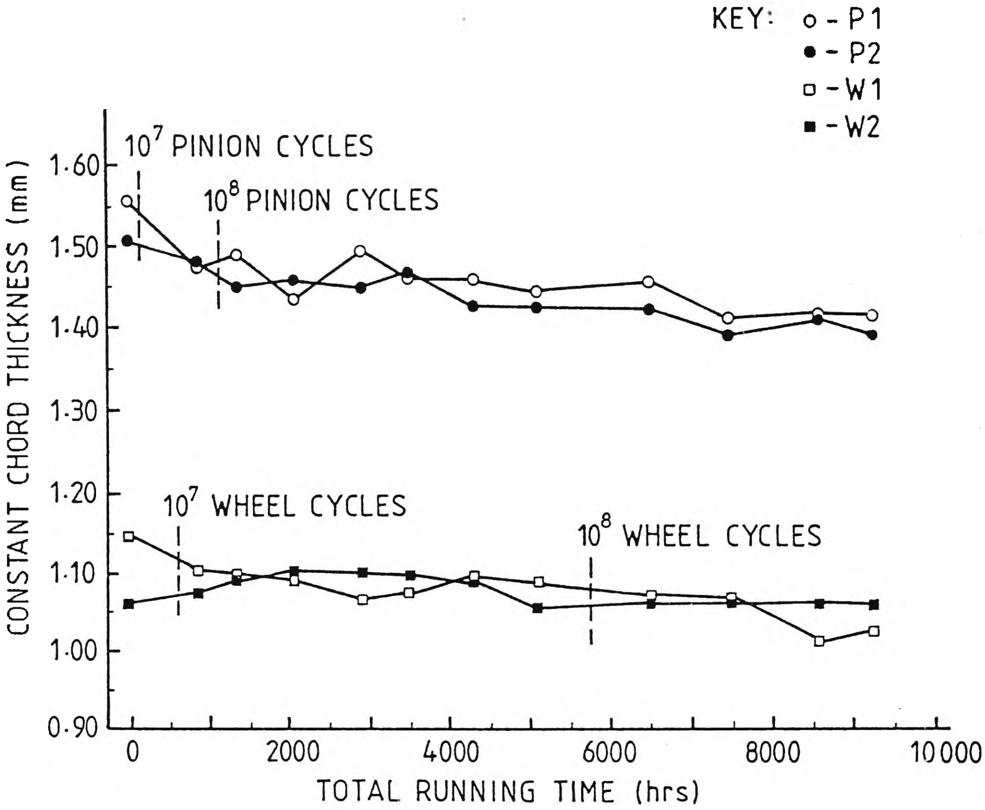


FIG. 8·3 WEAR RESULTS FOR GEAR SET 1

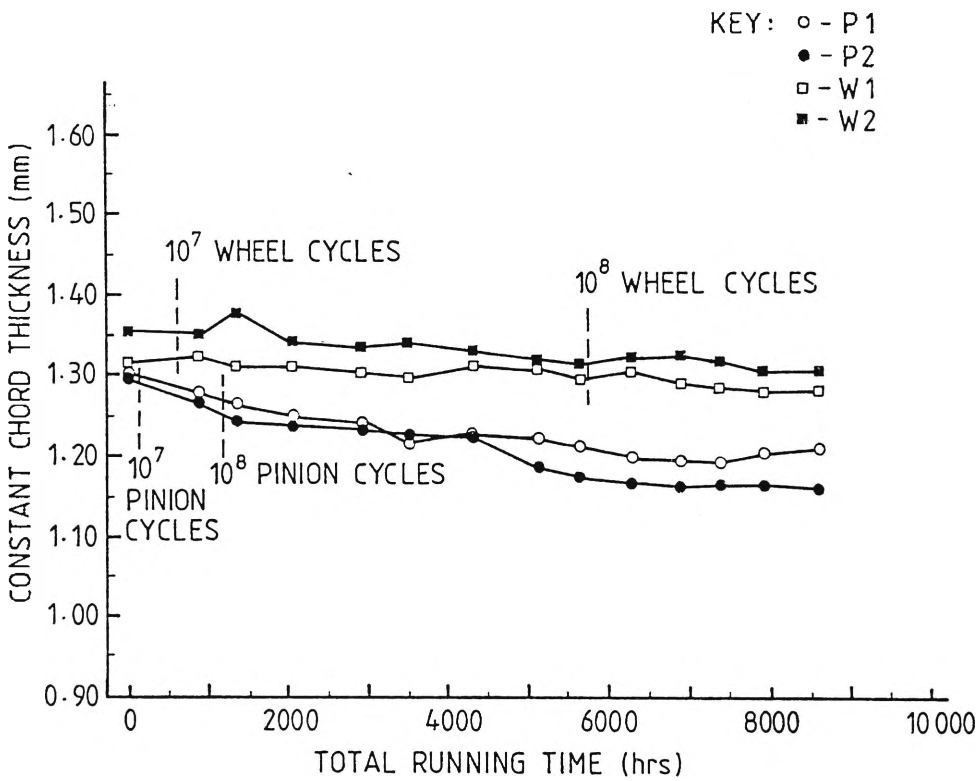


FIG. 8·4 WEAR RESULTS FOR GEAR SET 2

8.6 Experimental Results

The tooth thicknesses measured using the procedure outlined in the previous section are summarised in Figures 8.3 to 8.6 inclusive, whilst Figures 8.7 to 8.14 inclusive show the pictorial record.

The accuracy of the optical measurement of the chordal thickness would at times appear to be questionable. For example, in Figure 8.3, the measurement of the wheel tooth thickness at position 2 shows variations in chordal thickness that are of the same order of magnitude as the total measured wear. These variations were remeasured by the preparation of new tooth replicas which invariably verified the initial measurement. This observation led the author to the conclusion that apparent errors in measurement resulted from localised plastic deformation of the tips of the teeth which may have led to an error in the measurement of the chordal height and subsequently the chordal thickness.

It would be well to note that the pitch of the teeth of this test work border on the very fine end of those used for heavy duty industrial practices and that caution should be used in attempting to extrapolate findings to coarse pitch gearing.

From Tables 8.3 and 8.4 one may conclude, subject to qualification, that there is a reasonable correlation between the AGMA and ISO codes, both predicting approximately a 3 kW wear rating for gear sets 1, 2 and 3. This is to be compared with Table 8.2 which shows a BS wear rating of approximately 2 kW. None of the codes considered predicts any appreciable difference in the wear ratings of gear sets 1 and 2, although BS 436 recommends against the design utilised in gear set 2. From Figures 8.3, 8.4 and 8.5 it can be seen that the rate of wear as measured by tooth thickness is approximately equal for all pinions and wheels of gear sets 1, 2, and 3. This is in good agreement with theory as all the codes predicted approximately the same relative wear ratings for all pinions and wheels. However, whether the rating should be 2 kW or 3 kW cannot be ascertained at this point as the gears have operated for only 9,000 hours out of their total design life of 52,000 hours.

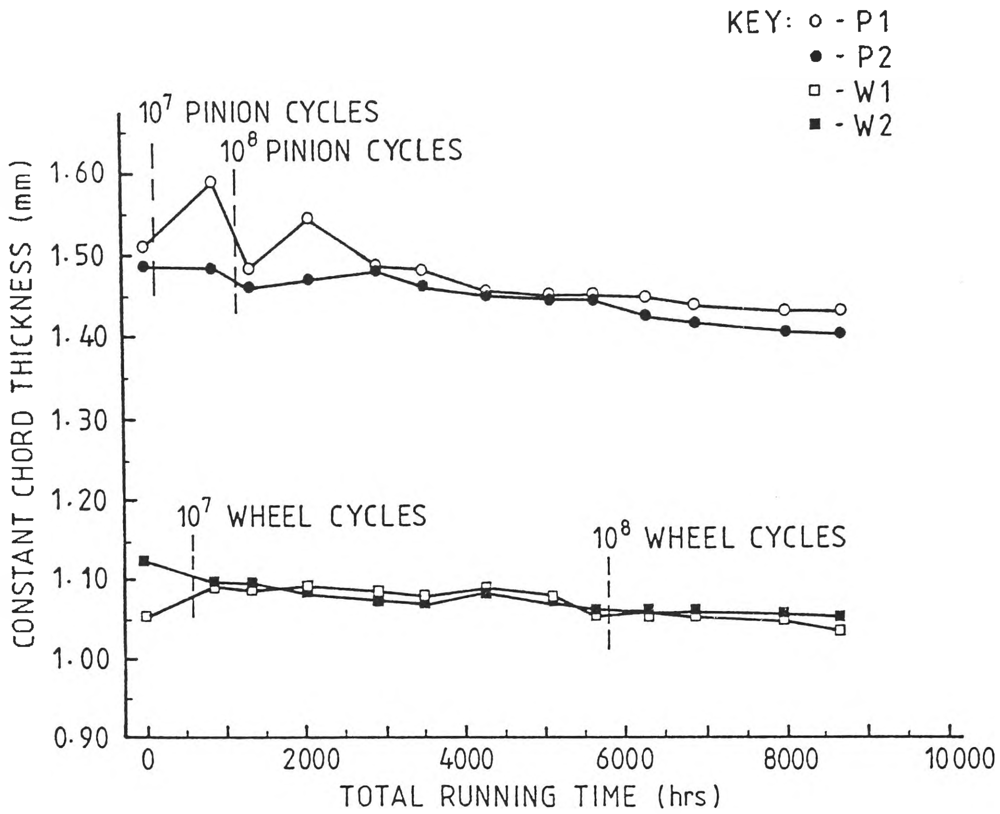


FIG. 8·5 WEAR RESULTS FOR GEAR SET 3

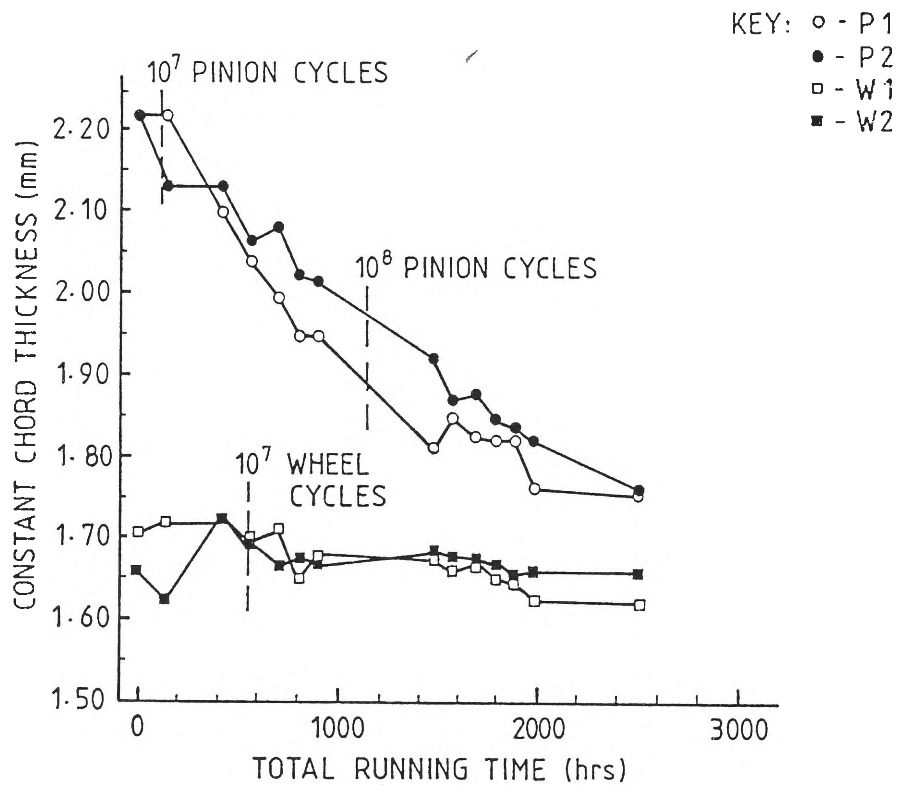


FIG. 8·6 WEAR RESULTS FOR GEAR SET 4

A visual inspection of the gear sets as shown in Figures 8.7 to 8.12 inclusive, suggests a different conclusion. Both the pinion and wheel of gear sets 1 and 2 are showing signs of pitting, the pitting being far more pronounced on gear set 2. The pinion of gear set 2 also exhibits a marked deformation of material along the top edges of the teeth. Gear set 3 is starting to show signs of slight pitting, but otherwise is in excellent condition. Many more hours of testing will be required to quantify the nature of the pitting, ie as to whether it will stabilise or continue to deteriorate. However, sufficient evidence exists to suggest that gear set 1 is a better design than gear set 2. The large number of teeth employed and the relatively small amount of addendum modification resulted in the original teeth shapes being similar and hence the question as to why gear set 1 has performed better than gear set 2 is presented. Each reader will have their own explanation. However, the author has conducted an extensive analysis of the involute geometry of each gear set and has concluded that the only significant variance is in the lengths of the approach and recess paths.

For gear set 1, the lengths of the approach and recess paths are 1.96mm and 3.44mm respectively, whilst for gear set 2 the same lengths are 2.86mm and 2.67mm respectively. This, in turn, has an effect on the calculated slide to roll ratio. Depending on the criteria employed, some gear designers consider that for a speed reducing gear pair, the length of the approach path should be less than the recess path with the slide to roll ratio being approximately equal for both the pinion and the wheel. Gear set 1 satisifies this criterion whilst gear set 2 does not. Without stating categorically that this is the only explanation for the difference in performance, let it suffice to say that none of the codes considered places restrictions on the ratio of the lengths of the approach and recess paths, even though both lengths are required in the calculation of geometry factors. However, it can be shown that the addendum modifications recommended by BS 436 usually lead to the length of the approach path being less than that of the recess path for a pair of speed reducing gears. The recommendations of ISO/TR 4467(7) also have a tendency to head in this direction.

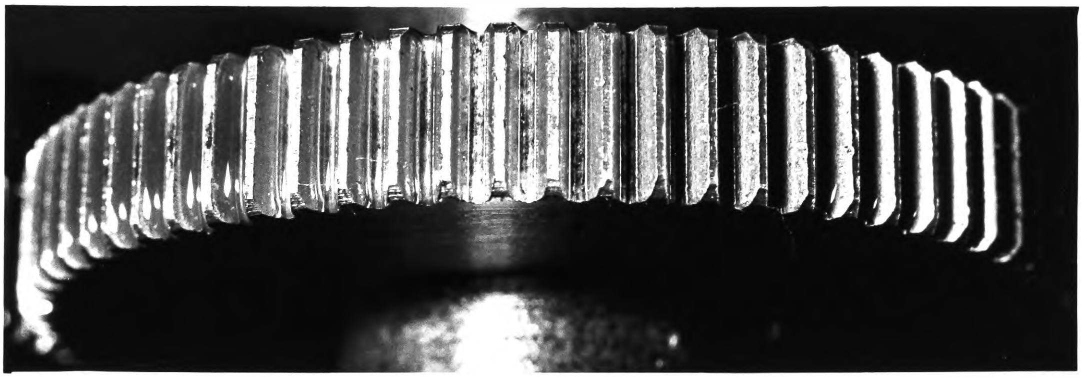


FIGURE 8.7 - TEETH OF GEAR SET 1 PINION - 9226 HOURS

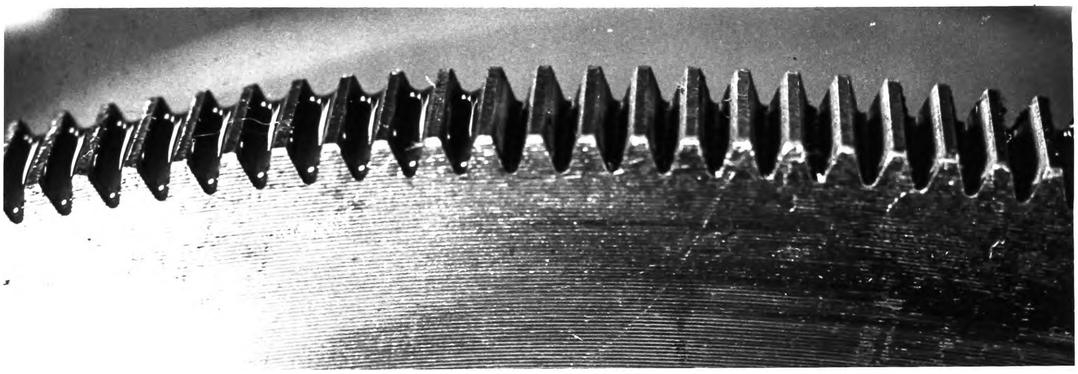


FIGURE 8.8 - TEETH OF GEAR SET 1 WHEEL - 9226 HOURS

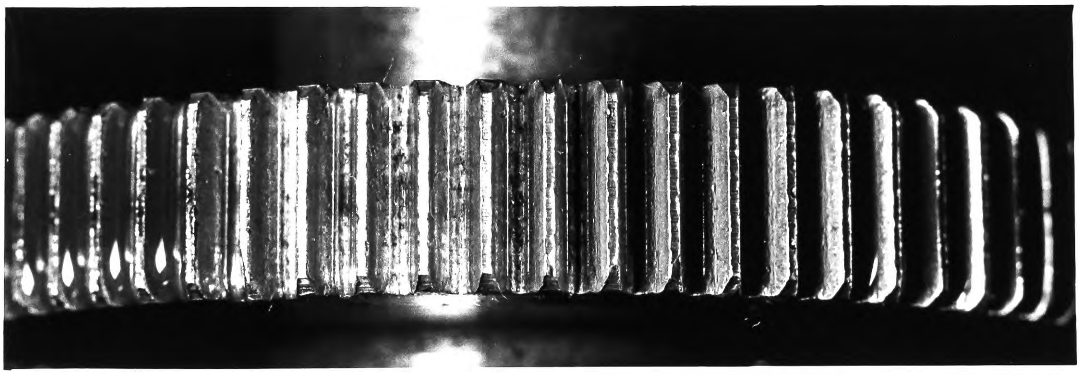


FIGURE 8.9 - TEETH OF GEAR SET 2 PINION - 8573 HOURS

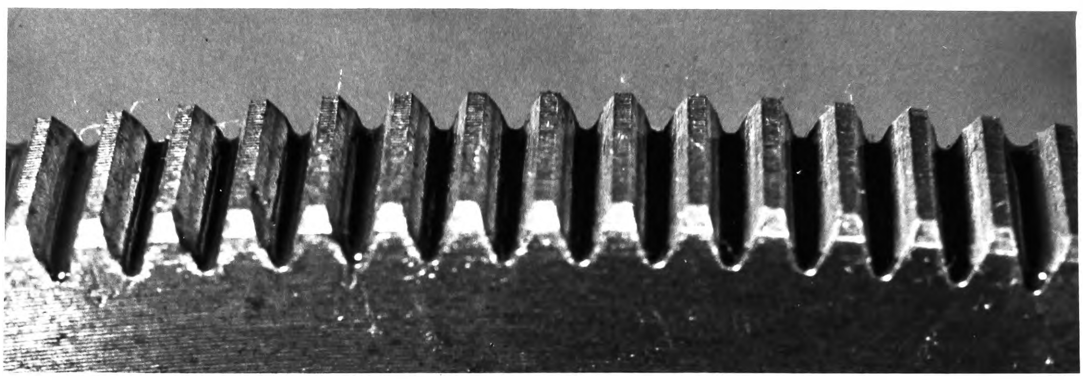


FIGURE 8.10 - TEETH OF GEAR SET 2 WHEEL - 8573 HOURS

Gear set 4 was originally designed for a BS 436 design life of 0.5 hours per day, corresponding to a total life of 1,083 hours or approximately 10^8 cycles for the pinion. From Figure 20 of AGMA 218.01 and Figure 12 of ISO 6336/2, this number of cycles has little effect on the life factor based on S - N curves. However, when one considers a design life of 1,083 hours as compared to 52,000 hours, BS 436 Chart 11 predicts a significant increase in the life factor and hence the wear rating. Conversely, if the AGMA power rating for gear set 4 were to be 2 kW, the life factor would have to be increased to 1.3, corresponding to a design life of one hour. Similarly, the ISO life factor would need to be 1.5 corresponding to 23 hours of total operation. From Figures 8.6, 8.13 and 8.14, it can be seen that gear set 4 failed after 2,521 hours of total operation. However, after 1,083 hours of operation, corresponding to the original design life of six (6) years, the gear set exhibited significant deterioration such that in a normal industrial environment, it would have been replaced. This is seen from an examination of Figures 8.15 to 8.18.

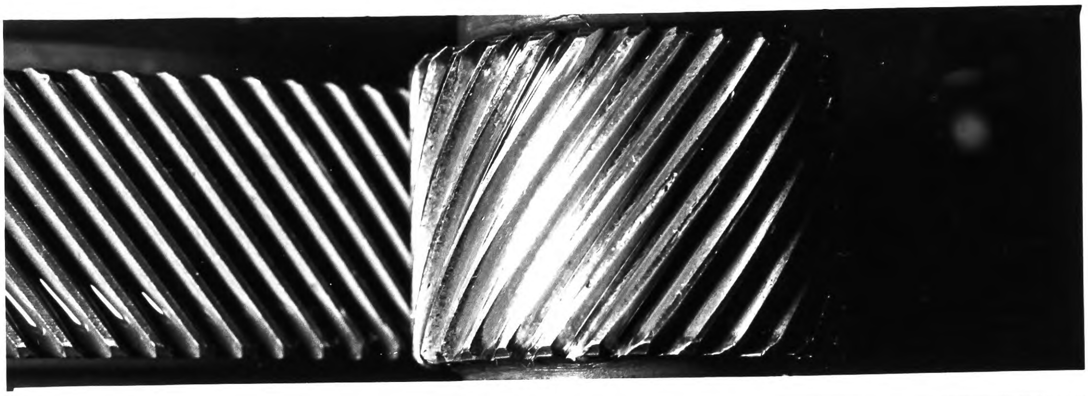


FIGURE 8.11 - TEETH OF GEAR SET 3 PINION - 8695 HOURS

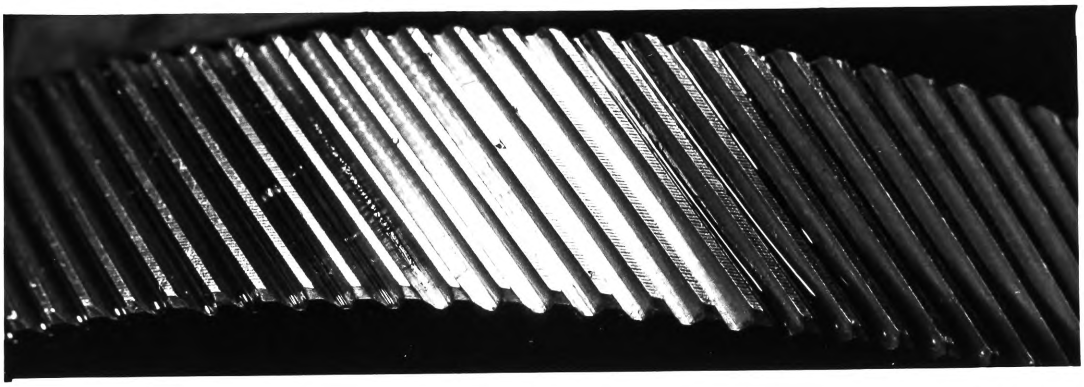


FIGURE 8.12 - TEETH OF GEAR SET 3 WHEEL - 8695 HOURS

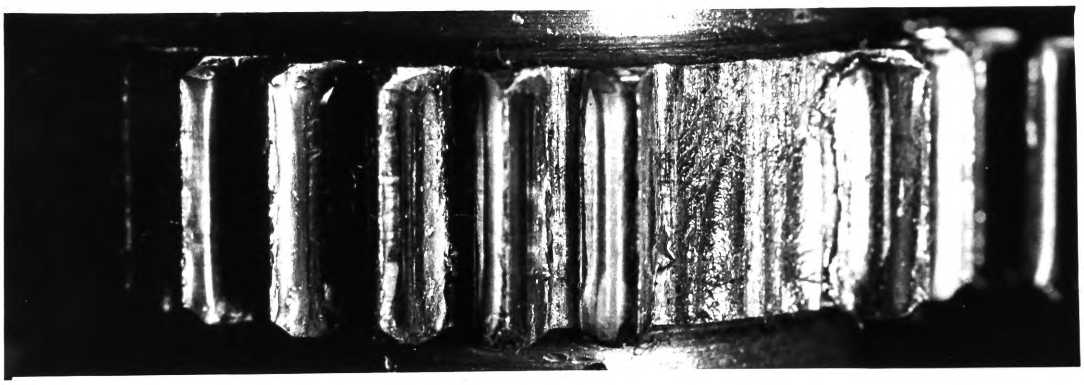


FIGURE 8.13 - TEETH OF GEAR SET 4 PINION - 2521 HOURS

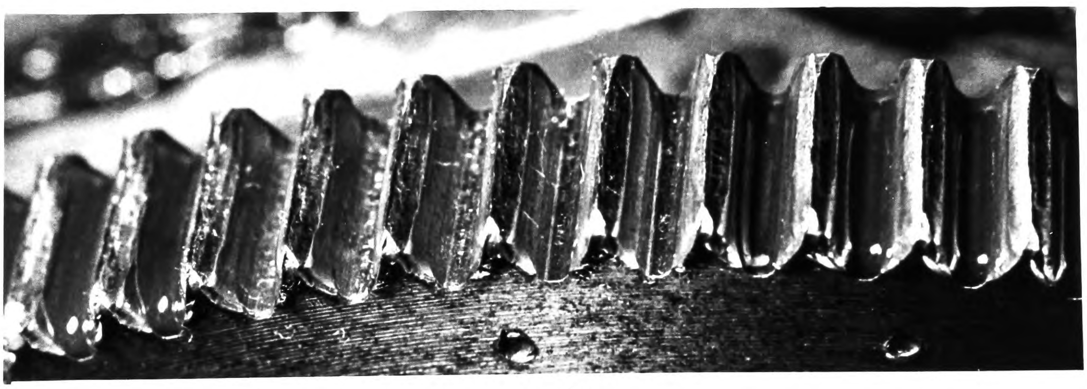


FIGURE 8.14 - TEETH OF GEAR SET 4 WHEEL - 2521 HOURS

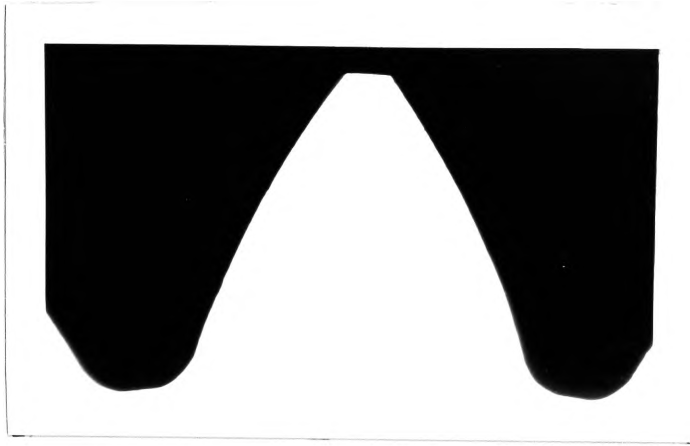


FIGURE 8.15 - PHOTOMICROGRAPH OF GEAR SET 4 PINION - 0 HOURS

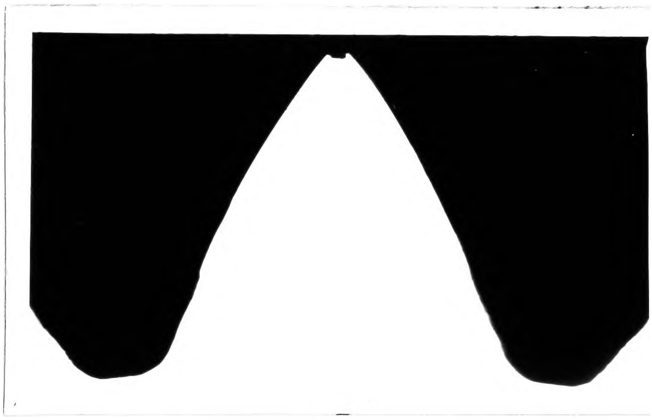


FIGURE 8.16 - PHOTOMICROGRAPH OF GEAR SET 4 PINION - 877 HOURS

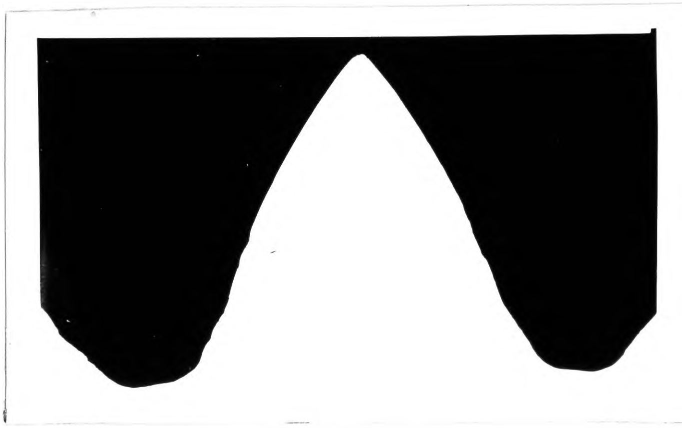


FIGURE 8.17 - PHOTOMICROGRAPH OF GEAR SET 4 PINION - 1601 HOURS

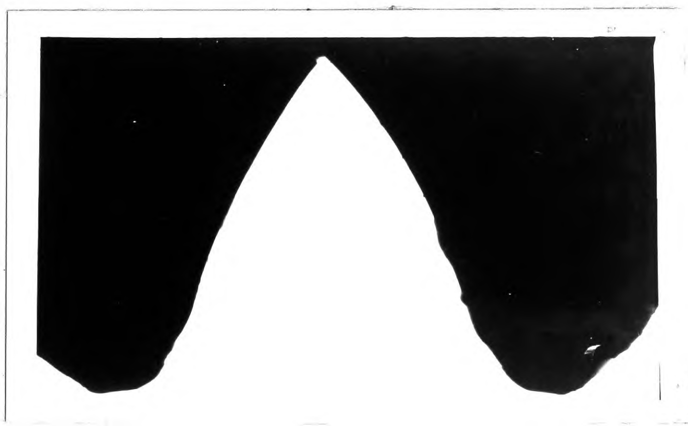


FIGURE 8.18 - PHOTOMICROGRAPH OF GEAR SET 4 PINION - 1869 HOURS

SECTION 9

CONCLUSIONS AND SUGGESTIONS FOR FURTHER STUDY

9.1 Conclusions

As stated in the synopsis, the tenet of this Thesis has been the presentation of an original analytical method for the calculation of the height and width of the Lewis parabola. Whilst this aim has been achieved, with the formal presentation constituting Section 1, the work has been amplified to the extent that thirty six graphs were produced for incorporation into AS 2938(5), together with the production of three pieces of software, namely GEARGEOM, IRATE and IDESIGN.

GEARGEOM was the first software program to be endorsed by the Standards Association of Australia, and accordingly has been designated as ASSP-001-1987(31).

As demonstrated in Section 2, the exact height and width of the Lewis equal strength parabola can be calculated and plotted by combining computer software and CAD (refer to Figures 2.8 to 2.13). However, it is the experience of the author that these facilities are not available to the majority of gear designers. This is especially true in tertiary institutions.

The gear power rating is directly proportional to the calculated geometry factor for bending strength, and hence errors in the manual drafting of the tooth profile reflect in the final gear design. When considering the wheel of many gear sets, the author has witnessed significant errors and frustration in students attempting to swing the large radii involved, to mention nothing of the time involved. Hence a need arose to produce design charts which would quickly yield the dimensions of the stress parabola. Figures 2.2 to 2.7 inclusive are charts which were developed in response to this need.

It is the author's opinion that the error introduced in calculating the equivalent number of teeth for conventional helical gears based on the helix angle at the standard pitch diameter is acceptable when compared with that calculated at the operating pitch diameter. Hence the stress parabola dimensions for a conventional helical gear can be read from Figures 2.2 and 2.3 using the equivalent number of teeth based on the helix angle at the standard diameter.

Further, the calculation of the load angle for spur and low contact ratio helical gears is commensurate with the time involved just to calculate the dimensions required for a tooth profile layout. Hence the time saving and the inherent improvement in accuracy is realised by the use of Figures 2.2 to 2.7 inclusive.

A major statistical analysis of the computer programs involved in the production of Figures 2.2 to 2.7 has shown that the order of accuracy is comparable with the errors introduced by not including backlash in the exact mathematical analysis. Hence, for any particular cutter, six charts can be produced which eliminate the need to manually draw a tooth profile to calculate the dimension of the stress parabola.

The values of J_P and J_G calculated in Section 3 using the approximation for the load sharing ratio are 0.537 and 0.545 respectively, whilst the exact mathematical solution, for a face contact ratio of 2, based on the actual load sharing ratio, yields 0.564 and 0.572 respectively.

Thus the values of J_P and J_G obtained using the charts represents errors of 5.0% and 4.7% respectively. However, these errors could be improved by a more sagacious choice of facewidth.

There are many gear applications which do not necessitate an optimisation of the design and as such a simple method to rate the gears would be advantageous.

The design charts presented in Section 4 complement the AGMA 218.01(4) system by providing the means of carrying out quick rating calculations.

The Standards Association of Australia realised the need for a quick rating procedure in introducing AGMA 218.01 as the new Australian Standard for the rating of spur and helical gears. In response to this requirement, twelve charts similar in format to those presented in Section 4 have been included in the Australian Standard AS 2938(5), thus facilitating an efficient rating procedure for numerous gear applications.

Section 5 has aimed to demonstrate that a significant improvement in power rating can be achieved by the correct choice of addendum modification coefficients, resulting in an improved gearbox at no additional cost to the manufacturer.

With the vehicle of a fully computerised analysis of geometry factors, the gear designer is now able to quantify what was previously instinctive feelings based on experience. Further, if the program incorporates adequate safeguards, addendum modification coefficients outside previously acceptable ranges can be considered.

The equations for the determination of the AGMA power ratings for a particular gear set are examined in Section 6. Each of the derating factors is examined in detail, and whilst most are expressed in a mathematical format in AGMA 218.01, several additional equations have been derived for the calculation of the life factor for pitting resistance, C_L, and the life factor for bending strength, K_L.

The allowable contact stress number, s_{ac}, and the allowable bending stress number, s_{at}, are in the opinion of the author, expressed in an ambiguous way in AGMA 218.01. The format in which allowable stress numbers are given in BS436-1940(21), would appear to be preferable, in that material specifications are expressed in commercial designations. An attempt has been made to demonstrate an apparent correlation between known material specifications and s_{ac} values.

The involute gear geometry associated with gear design constitutes Section 7. Particular attention has been directed towards the slide to roll ratio, a point of design which arguably has not been considered in AGMA 218.01, as opposed to DIN or ISO standards which consider a third mode of gear failure, namely that of scoring or scuffing.

A hypothesis has been presented for the modification of the AGMA 218.01 equation for the geometry factor for pitting resistance, I, such that it bears a correlation to the slide to roll ratio.

Experimental results obtained from the University of Wollongong's gear testing rig are highlighted in Section 8. The conclusions drawn from these results are summarised below.

Although four gear sets do not provide sufficient evidence from which to draw categorical conclusion, there are perhaps trends which can be highlighted.

Firstly, the power ratings as predicted by national codes bear a reasonable correlation to the actual power transmission capabilities of the gear sets tested. However, it appears that insufficient attention is drawn to a suitable choice of addendum modification coefficients, and in this regard it is suggested that AGMA consider either the adoption of ISO/TR 4467(7) or the production of a similar document.

Secondly, the design life based on the number of cycles of operation would appear to be conservative when one considers gear sets designed for a limited number of hours of operation at a relatively high speed of rotation. It would appear that a life factor based on speed of rotation and design life may be more appropriate than the current practice.

9.2 Suggestions for Further Study

Whilst a considerable number of man hours has been expended in the production of this Thesis over the past ten years, there are areas in which further study is recommended.

The "proof" of the convergence of the Lambda Method as depicted by Figure 1.9, is confined to the conventional limits of ISO/TR4467. A formal mathematical proof and/or an examination of convergence beyond the limits of ISO/TR4467 should be considered.

The technique which has been presented in Section 7, for a correlation between the geometry factor for pitting resistance, I, and the slide to roll ratios, S_P and S_G, was confined to spur gears at standard centres. This technique should be examined for spur gears at extended centres, together with helical gears at both standard and extended centre distances. Further, the gear test rig should be utilised to incorporate gear sets which are identical in all respects, with the exception of the addendum modification, to verify the pitting resistance as a function of the slide to roll ratio, as shown in Figures 7.12 and 7.15.

It is envisaged that this study would be an extension of the data previously collected for gear sets 1 and 2.

A theoretical analysis of the DIN/ISO standard to examine the correlation of the wear and scuffing power ratings as a function of the slide to roll ratio, would add to the understanding of this particular phenomenon.

Whilst BS PD6457 Nov 1970(15) provides an approximate solution to equation (7.10), the determination of the addendum modification coefficients to achieve an equal slide to roll ratio for both the pinion and wheel, warrants further attention.

SECTION 10

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APPENDIX A

COMPUTER PROGRAMME FOR CALCULATING GEOMETRY FACTORS I AND J

```
      SUBROUTINE ANAL
$INCLUDE: 'PC2.FOR'
C ****************************************************************
C * Version 1.03    AUSTRALIAN  STANDARD  APPENDIX  D              *
C * THIS DOES NOT FORM PART OF THE AUSTRALIAN STANDARD  AS2938-1987. *
C ****************************************************************
C *                                                              *
C *  CALCULATION OF THE GEOMETRY FACTORS FOR PITTING RESISTANCE AND *
C *    BENDING STRENGTH FOR EXTERNAL SPUR AND HELICAL INVOLUTE GEAR *
C *    TEETH AS PER AGMA 218.01 DEC.1982 & AS2938-1987             *
C ****************************************************************
C *                                                              *
C * ENTER THE DATA IN THE FOLLOWING ORDER.                        *
C * NOTE:- DEGREES ARE ENTERED AS FOLLOWS.                        *
C * 16 DEGREES 37 MINUTES 58 SECONDS IS ENTERED AS 16.3758 DEGREES. *
C *                                                              *
C ****************************************************************
C * VARIABLE   * UNITS * DESCRIPTION                              *
C ****************************************************************
C *            *       *                                          *
C * TITLE      *       * DESCRIPTION OF GEAR SET                  *
C * NT(1)(NP)  *       * NUMBER OF PINION TEETH                   *
C * NT(2)(NW)  *       * NUMBER OF WHEEL  TEETH                   *
C * PHIC       *  DEG  * NORMAL PROFILE ANGLE OF EQUIVALENT RACK CUTTER*
C * MN         *  MM   * NORMAL METRIC MODULE                     *
C * PSIS       *  DEG  * HELIX ANGLE AT STANDARD PITCH DIAMETER   *
C * X(1)(XP)   *       * ADDENDUM MODIFICATION COEFFICIENT OF PINION *
C * X(2)(XG)   *       * ADDENDUM MODIFICATION COEFFICIENT OF WHEEL  *
C * F          *  MM   * NET FACE WIDTH OF NARROWEST MEMBER       *
C * HA(1)      *  MM   * STANDARD ADDENDUM OF THE BASIC RACK FOR PINION*
C * HA(2)      *  MM   * STANDARD ADDENDUM OF THE BASIC RACK FOR WHEEL *
C * HB(1)      *  MM   * STANDARD DEDENDUM OF THE BASIC RACK FOR PINION*
C * HB(2)      *  MM   * STANDARD DEDENDUM OF THE BASIC RACK FOR WHEEL *
C * RT(1)      *  MM   * EDGE RADIUS OF CUTTING TOOL FOR PINION   *
C * RT(2)      *  MM   * EDGE RADIUS OF CUTTING TOOL FOR WHEEL    *
C * DELTAO(1)  *  MM   * PROTUBERANCE OF CUTTING TOOL FOR PINION  *
C * DELTAO(2)  *  MM   * PROTUBERANCE OF CUTTING TOOL FOR WHEEL   *
C * DLTARO(1)  *  MM   * TRUNCATION APPLIED TO PINION             *
C * DLTARO(2)  *  MM   * TRUNCATION APPLIED TO WHEEL              *
C * BN(1)      *  MM   * BACKLASH APPLIED TO PINION               *
C * BN(2)      *  MM   * BACKLASH APPLIED TO WHEEL                *
C * ICODE      *       * 0 = ITERATION TO CODE FOR LAMBDA         *
C *            *       * 1 = FULL ITERATION FOR LAMBDA            *
C * ROUGH      *       * 0 = ACCURATE SPUR GEARS                  *
C *            *       * 1 = INACCURATE SPUR GEARS                *
C * BUTT(1)    *       * 0 = NO PINION BUTTRESS   1 = PINION BUTTRESS *
C * BUTT(2)    *       * 0 = NO WHEEL BUTTRESS    1 = WHEEL BUTTRESS  *
C *            *       *                                          *
C ****************************************************************
C
      DOUBLE PRECISION ANH, ANI, ANJ, ANK, ANL, ANO, ANP, CX, CXH
      DOUBLE PRECISION CPSI, CC, LAMBDI, CRACK, CT
      DOUBLE PRECISION I, INV
      DOUBLE PRECISION J(2)
      DOUBLE PRECISION KF(2), KPSI(2), K1(2), K2(2), K3(2), K4(2)
```

```
      DOUBLE PRECISION K5(2), K6(2), K7(2), K8(2), K9(2)
      DOUBLE PRECISION L(2), LAMBDA(2), LAMBAI(2), LAMBA1(2), LMIN
      DOUBLE PRECISION M(2), MF, MNN, MP, MT
      DOUBLE PRECISION NE(2), NT(2)
      DOUBLE PRECISION A(2), AB(2), ANC(2), ANCRF(2), ANN(2), ANM(2)
      DOUBLE PRECISION B(2), BETA(2), BETAH(2), BETARH(2), BIGRF(2)
      DOUBLE PRECISION CH(2), CI(2), CM(2), CN(2), CU(2), CW(2)
      DOUBLE PRECISION D(2), DR(2), DS(2), D0(2), TDLTAR(2)
      DOUBLE PRECISION DLTAR3(2), DLTAR4(2), DLTAR5(2)
      DOUBLE PRECISION EPSLON(2)
      DOUBLE PRECISION FMIN(2)
      DOUBLE PRECISION H(2), HAB(2), HE(2)
      DOUBLE PRECISION PHIL(2), PHILN(2), PHIN(2), PHIO(2), PSI(2)
      DOUBLE PRECISION PSIO(2), PSIR(2), PHIR(2), TPHIC, TPSIS
      DOUBLE PRECISION PHIT, PHITI
      DOUBLE PRECISION R(2), RB(2), RBE(2), RE(2), RF(2), RMID(2)
      DOUBLE PRECISION ROE(2), ROM(2), RR(2), RS(2), RU(2), RO(2)
      DOUBLE PRECISION SMALLC(2), SPAN(2), STEETH(2)
      DOUBLE PRECISION TE(2), TNC(2), TO(2), TOT(2), TST(2), TRT(2)
      DOUBLE PRECISION Y(2)
      DOUBLE PRECISION ZA, ZB,ZC, ZCH, ZNE
      CHARACTER*55 OU(4), OUT
      CHARACTER*42 OT(4), OTT
      CHARACTER*19 OS(2), OSS
      CHARACTER*14 TYPE(2)
      CHARACTER*10 YN(2), YP, YW, Y1, Y2, GP, GW, LP, LW, MTYES
      LOGICAL SPUR, LCR, HELIC, OUTPUT, PINION, WHEEL, UNDERP, UNDERW
      LOGICAL GAPP, GAPW, LANDP, LANDW, NONCON
      INTEGER ITEETH(2), NWT, CNT1, CNT2
      DATA OU/' GEOMETRY FACTOR - ACCURATE SPUR GEARS - AS2938-1987. ',
     *        ' GEOMETRY FACTOR - INACCURATE SPUR GEARS - AS2938-1987.',
     *        'GEOMETRY FACTOR-CONVENTIONAL HELICAL GEARS-AS2938-1987.',
     *        '    GEOMETRY FACTOR - LCR HELICAL GEARS - AS2938-1987. '/
      DATA OT/'       ACCURATE SPUR GEARS TO AS2938-1987.',
     *        '     INACCURATE SPUR GEARS TO AS2938-1987.',
     *        'CONVENTIONAL HELICAL GEARS TO AS2938-1987.',
     *        '          LCR HELICAL GEARS TO AS2938-1987.'/
      DATA OS/' MODIFIED ADDENDA  ' , ' UNMODIFIED ADDENDA' /
      DATA YN     / '   YES    ', '    NO    ' /
      DATA TYPE   / 'CODE ITERATION' , 'FULL ITERATION' /
C
C     STATEMENT FUNCTIONS.
C     ===================
C
C     CONVERT RADIANS TO DECIMAL DEGREES
      RTD(SFA) = SFA * 180.0 / PI
C     CONVERT DECIMAL DEGREES TO RADIANS
      DTR(SFB) = SFB * PI / 180.0
C     CONVERT SECONDS TO DECIMAL OF DEGREES
      SE(SFC)  = INT( ( SFC*100.0 - INT(SFC*100.0) )*100.0) / 3600.0
C     CONVERT MINUTES TO DECIMAL OF DEGREES
      SF(SFD)  = INT( ( SFD-INT(SFD) )*100.0 )/60.0
C     OBTAIN DECIMAL DEGREES FROM DEGREES.MINUTES,SECONDS
      DGS(SFE) = INT(SFE) + SF(SFE) + SE(SFE)
C     OBTAIN SECONDS FROM DECIMAL DEGREES
      NR(SFF)  = ( (SFF-INT(SFF))*60.0 - INT( (SFF-INT(SFF))*60.0 ) )
     *           * 60.0 + 0.5
```

```
C       OBTAIN MINUTES FROM DECIMAL DEGREES
        NS(SFG)  = ( SFG -INT(SFG) )*60.0
C       OBTAIN WHOLE DEGREES FROM DECIMAL DEGREES
        NMS(SFH) = INT(SFH)
C       OBTAIN WHOLE DEGREES FROM DEGREES.MINUTES,SECONDS
        NX(SFJ)  = INT(SFJ)
C       OBTAIN MINUTES FROM DEGREES.MINUTES,SECONDS
        NY(SFK)  = INT( ( SFK - INT(SFK) ) * 100.0 )
C       OBTAIN SECONDS FROM DEGREES.MINUTES,SECONDS
        NZ(SFL) = INT( ( SFL*100.0 - INT(SFL*100.0) ) * 100.0 )
C       OBTAIN FRACTIONAL PART OF A NUMBER
        FRAC(SFM) = SFM - INT(SFM)
C       OBTAIN INVOLUTE OF AN ANGLE
        INV(SFO) = TAN(SFO) - SFO
C
C       STORE ORIGINAL DLTARO, PHIC AND PSIS FOR RETURN
C
        TDLTAR(1) = DLTARO(1)
        TDLTAR(2) = DLTARO(2)
        TPHIC = PHIC
        TPSIS = PSIS

C       INITIALIZE GEAR SET NUMBER AND CALCULATE PI.
C       ===========================================
C
        PI = 4.0 * ATAN(1.0)
C
        DLTAR3(1) = 0.0
        DLTAR3(2) = 0.0
        DLTAR4(1) = 0.0
        DLTAR4(2) = 0.0
        DLTAR5(1) = 0.0
        DLTAR5(2) = 0.0
        OLDDRO(1) = 0.0
        OLDDRO(2) = 0.0
C
C
C
        IF ( IOPT .GT. 1 ) WRITE ( 6, 2002 )
2002    FORMAT ( 1H1, \    )
        WRITE ( 6, 21 ) NN, TYPE(ICODE+1), TITLE
   21   FORMAT ( 9H GEAR SET, I3, 5X, A14 / 1X, A68  )
        NT(1) = NP
        NT(2) = NW
C       ROUNDING UP TO NEAREST SECOND
CCCC    PSIS = PSIS + 0.00005
CCCC    PHIC = PHIC + 0.00005
        AN1 = PHIC
        AN2 = PSIS
C
C       DETERMINE TYPE OF GEAR AND PRINT HEADING
C       ========================================
C
C       CONVERT DEGREES TO RADIANS
C         FROM DEG,MIN,SEC TO DECIMAL DEGREES
        PSIS = DGS(PSIS)
        PHIC = DGS(PHIC)
```

```
C         FROM DECIMAL DEGREES TO RADIANS
      PSIS = DTR(PSIS)
      PHIC = DTR(PHIC)
      MF = F * SIN(PSIS) / ( PI * MN )
C     SPUR GEARS
      SPUR  = PSIS .EQ. 0.0
C     CONVENTIONAL HELICAL GEARS
      HELIC = MF .GT. 1.0 .AND..NOT. SPUR
C     LOW CONTACT RATIO HELICAL GEARS
      LCR   = MF .LE. 1.0 .AND..NOT. SPUR
      IF ( SPUR  ) THEN
        OUT =  OU(1)
        OTT =  OT(1)
      ENDIF
      IF ( ROUGH .EQ. 1 ) THEN
        OUT =  OU(2)
        OTT =  OT(2)
      ENDIF
      IF ( HELIC ) THEN
        OUT =  OU(3)
        OTT =  OT(3)
      ENDIF
      IF ( LCR   ) THEN
        OUT =  OU(4)
        OTT =  OT(4)
      ENDIF
      OSS = OS(1)
      IF ( X(1) .EQ. 0.0 .AND. X(2) .EQ. 0.0 ) OSS = OS(2)
      WRITE ( 6, 829 ) OUT
      WRITE ( 6, 830 ) NP, NW, MN, MN,
     * NX(AN1), NY(AN1), NZ(AN1), NX(AN1), NY(AN1), NZ(AN1),
     * HA, HB, RT, DELTAO,
     * NX(AN2), NY(AN2), NZ(AN2), NX(AN2), NY(AN2), NZ(AN2)
  829 FORMAT ( 1H , A55, / 1H , 78(1H*), /
     *1H ,  '*STEP* NAME   *   PINION   *    WHEEL   *UNIT*',
     * ' DESCRIPTION', 19X, 1H*, / 1H , 78(1H*) )
      WRITE ( 6, 831 ) F, F, X, DLTARO, BN
      IF ( HELIC ) WRITE ( 6, 833 ) BUTT
      WRITE ( 6, 834 ) QV
  830 FORMAT (
     *   ' *   I * NP,NW  *', 2 (I11, 2H *),
     *   '     * NUMBER OF TEETH                 *' /
     *   ' *   N * MN     *', 2 (F11.3, 2H *),
     *   ' MM * NORMAL METRIC MODULE             *' /
     *   ' *   P * PHIC   *', 2 (I4,1X,I2.2,1H',I2.2,1H", 2H *),
     *   ' DEG* NORMAL PRESSURE ANGLE            *' /
     *   ' *   U * HA     *', 2 (F11.3, 2H *),
     *   ' MM * STANDARD ADDENDUM OF TOOL        *' /
     *   ' *   T * HB     *', 2 (F11.3, 2H *),
     *   ' MM * STANDARD DEDENDUM OF TOOL        *' /
     *   ' *     * RT     *', 2 (F11.3, 2H *),
     *   ' MM * TIP RADIUS OF CUTTING TOOL       *' /
     *   ' *     * DELTAO *', 2 (F11.3, 2H *),
     *   ' MM * PROTUBERANCE OF CUTTING TOOL     *' /
     *   ' *   D * PSIS   *', 2 (I4,1X,I2.2,1H',I2.2,1H", 2H *),
     *   ' DEG* HELIX ANGLE AT REF PITCH DIA     *' )
  831 FORMAT (
```

```
     *  ' *  A * F      *', 2 (F11.3, 2H *),
     *  ' MM * NET FACE WIDTH                   *' /
     *  ' *  T * X      *', 2 (F11.3, 2H *),
     *  '    * ADDENDUM MODIFICATION COEFF      *' /
     *  ' *  A * DLTARO *', 2 (F11.3, 2H *),
     *  ' MM * TRUNCATION APPLIED               *' /
     *  ' *    * BN     *', 2 (F11.3, 2H *),
     *  ' MM * BACKLASH APPLIED                 *' )
  833 FORMAT (
     *  ' *    * BUTT   *', 2 (I8,3X, 2H *),
     *  '    * BUTTRESSING (0=NO, 1=YES)        *'  )
  834 FORMAT (
     *  ' *    * QV     *', 2 (I8,3X, 2H *),
     *  '    * AGMA 390 QUALITY NUMBER          *'  /
     * 1H , 78(1H*) )
      IF ( IOPT .EQ. 1 ) CALL PAUS
      WRITE ( 6, 835 )
      WRITE ( 6, 829 ) OUT
  835 FORMAT (    ' INVOLUTE GEAR MATHEMATICS' )
C
C
C     CALCULATE 'I' AND 'J' GEOMETRY FACTORS FOR GEAR SET
C     ===================================================
C
C
C
C     STEP 1      PHIS = TRANSVERSE PRESSURE ANGLE
C     =======
C

      PHIS = ATAN ( TAN(PHIC) / COS(PSIS) )
      ANG = RTD(PHIS) + 0.00005
C
C     STEP 2      PHIT = OPERATING TRANSVERSE PRESSURE ANGLE
C     =======
C
      IF ( X(1) + X(2) .NE. 0.0 ) GO TO 2
      PHIT = PHIS
      GO TO 152
    2 PHITI = 2.0 * ( X(1) + X(2) ) * TAN(PHIC)
     *       / ( NT(1) + NT(2) ) + INV(PHIS)
      IF ( ICODE .NE. 0 ) GO TO 52
C
C     CODE ITERATION FOR PHIT
C
      PHI1 = PHIS - ( ( INV(PHIS) - PHITI ) / ( TAN(PHIS) ** 2 ) )
      PHI2 = PHI1 - ( ( INV(PHI1) - PHITI ) / ( TAN(PHI1) ** 2 ) )
      PHIT = PHI2 - ( ( INV(PHI2) - PHITI ) / ( TAN(PHI2) ** 2 ) )
      GO TO 152
C
C     FULL ITERATION FOR PHIT  ACTIVATED BY ICODE = 1
C
   52 PHI1 = PHIS
  102 PHI1 = PHI1 - ( ( INV(PHI1) - PHITI ) / ( TAN(PHI1) ** 2 ) )
      IF ( ABS( INV(PHI1) - PHITI ) .GE. 1.0E-08 ) GO TO 102
      PHIT = PHI1
  152 CONTINUE
```

```
      ANH = RTD(PHIT) + 0.00005
C
C     STEP 3      C      = OPERATING CENTRE DISTANCE
C     =========
C
      C = 0.5 * MN * ( NT(1) + NT(2) ) * COS(PHIS)
     *  /  ( COS(PSIS) * COS(PHIT) )
C
C     STEP 4      RB(1) = BASE RADIUS PINION
C     =======     RB(2) = BASE RADIUS WHEEL
C
      RB(1) = 0.5 * NT(1) * MN * COS(PHIS) / COS(PSIS)
      RB(2) = RB(1) * NT(2) / NT(1)
C
C     STEP 5      R(1)    = OPERATING PITCH RADIUS PINION
C     =======     R(2)    = OPERATING PITCH RADIUS WHEEL
C
      R(1) = NT(1) * C / ( NT(1) + NT(2) )
      R(2) = C - R(1)
C
C     STEP 6      RO(1)   = TIP RADIUS PINION
C     =======     RO(2)   = TIP RADIUS WHEEL
C
      RO(1) = MN * ( ( 0.5 * NT(1) / COS(PSIS) ) + X(1) )
     *        + HB(1) - DLTARO(1)
      RO(2) = MN * ( ( 0.5 * NT(2) / COS(PSIS) ) + X(2) )
     *        + HB(2) - DLTARO(2)
C
C     STEP 6A
C     =======
C
C     *****************************************************************
C     *CHECK TIP-FILLET INTERFERENCE, TIP THICKNESS AND CONJUGATE ACTION*
C     *  THE FOLLOWING CALCULATIONS ARE NECESSARY TO CALCULATE THE      *
C     *  THE RADIUS TO THE TOP OF THE TROCHOID AND THE TOP LAND WIDTH   *
C     *  REF:- EARLE BUCKINGHAM  --  MANUAL OF GEAR DESIGN ( SECT 3 )   *
C     *        INDUSTRIAL PRESS INC. NEW YORK.                          *
C     *****************************************************************
C
C     STEP 6A.1  CHECK FOR UNDERCUT TEETH
C     =========
C
C                RS(1)     = REFERENCE PITCH RADIUS PINION
C                RS(2)     = REFERENCE PITCH RADIUS WHEEL
C                SMALLC(1)= CUTTER TIP RADIUS CONSTANT PINION
C                SMALLC(2)= CUTTER TIP RADIUS CONSTANT WHEEL
C                RU(1)     = UNDERCUT RADIUS PINION
C                RU(2)     = UNDERCUT RADIUS WHEEL
C                RR(1)     = ROOT RADIUS PINION
C                RR(2)     = ROOT RADIUS WHEEL
C                BIGRF(1) = RADIUS TO TOP OF TROCHOID PINION
C                BIGRF(2) = RADIUS TO TOP OF TROCHOID WHEEL
C
      RS(1) = 0.5 * NT(1) * MN / COS(PSIS)
      RS(2) = 0.5 * NT(2) * MN / COS(PSIS)
      SMALLC(1) = RT(1) * ( 1 - SIN(PHIC) )
      SMALLC(2) = RT(2) * ( 1 - SIN(PHIC) )
```

```
      RU(1) = RS(1) * COS(PHIS) ** 2 - SMALLC(1)
      RU(2) = RS(2) * COS(PHIS) ** 2 - SMALLC(2)
      RR(1) = RS(1) - HA(1) + MN * X(1) - ( 0.5 * BN(1) / SIN(PHIC))
      RR(2) = RS(2) - HA(2) + MN * X(2) - ( 0.5 * BN(2) / SIN(PHIC) )
      UNDERP = RR(1) .LT. RU(1)
      UNDERW = RR(2) .LT. RU(2)
      IF ( UNDERP ) THEN
        Y1 = YN(1)
        BIGRF(1) = RB(1) + ( RU(1) - RR(1) ) ** 2
     *             / ( 6.0 * RS(1) * COS(PHIS) * SIN(PHIS) ** 2 )
      ELSE
        Y1 = YN(2)
        BIGRF(1) = SQRT ( ( ( RS(1) - RR(1) - SMALLC(1) )
     *             / TAN(PHIS) ) ** 2
     *             + ( RR(1) + SMALLC(1) ) ** 2 )
      ENDIF
      IF ( UNDERW ) THEN
        Y2 = YN(1)
        BIGRF(2) = RB(2) + ( RU(2) - RR(2) ) ** 2
     *             / ( 6.0 * RS(2) * COS(PHIS) * SIN(PHIS) ** 2 )
      ELSE
        Y2 = YN(2)
        BIGRF(2) = SQRT ( ( ( RS(2) - RR(2) - SMALLC(2) )
     *             / TAN(PHIS) ) ** 2
     *             + ( RR(2) + SMALLC(2) ) ** 2 )
      ENDIF
      WRITE ( 6, 1110 ) RS, SMALLC, RU, RR, Y1, Y2,
     *                     BIGRF, RO
 1110 FORMAT (
     * ' * 6A * RS      *', 2 (F11.3, 2H *),
     * ' MM * REFERENCE PITCH RADII          *' /
     * ' * 6A * SMALLC *', 2 (F11.3, 2H *),
     * ' MM * CUTTER TIP RADII CONSTANT      *' /
     * ' * 6A * RU      *', 2 (F11.3, 2H *),
     * ' MM * UNDERCUT RADII                 *' /
     * ' * 6A * RR      *', 2 (F11.3, 2H *),
     * ' MM * ROOT RADII                     *' /
     * ' * 6A *         *', 2 (A11,    2H *),
     * '      * UNDERCUT (YES=RU>RR)         *'  /
     * ' * 6A * BIGRF   *', 2 (F11.3, 2H *),
     * ' MM * RADII TO TOP OF TROCHOID       *' /
     * ' * 6  * RO      *', 2 (F11.3, 2H *),
     * ' MM * ORIGINAL TIP RADII             *' )
C
C     STEP 6A.2   CHECK FOR TIP TO TROCHOID INTERFERENCE
C     =========
C
C                 ROM(1)    = MAXIMUM ALLOWABLE TIP RADIUS PINION
C                 ROM(2)    = MAXIMUM ALLOWABLE TIP RADIUS WHEEL
C                 CI(1)     = INVOLUTE CLEARANCE COEFFICIENT PINION
C                 CI(2)     = INVOLUTE CLEARANCE COEFFICIENT WHEEL
C                 CIREF = REFERANCE INVOLUTE CLEARANCE COEFFICIENT
C
      ROM(1) = SQRT ( ( C * SIN(PHIT)
     *        - SQRT ( BIGRF(2) ** 2 - RB(2) ** 2 ) ) ** 2 + RB(1) ** 2 )
     *        + 0.001
      ROM(2) = SQRT ( ( C * SIN(PHIT)
```

```
     *         - SQRT ( BIGRF(1) ** 2 - RB(1) ** 2 ) ) ** 2 + RB(2) ** 2 )
     *         + 0.001
      CI(1)  = ( SQRT ( ( C * SIN(PHIT)
     *         - SQRT ( RO(1) ** 2 - RB(1) ** 2 ) ) ** 2 + RB(2) ** 2 )
     *         - BIGRF(2) ) / MN
      CI(2)  = ( SQRT ( ( C * SIN(PHIT)
     *         - SQRT ( RO(2) ** 2 - RB(2) ** 2 ) ) ** 2 + RB(1) ** 2 )
     *         - BIGRF(1) ) / MN
      PINION = CI(1) .LT. CIREF
      WHEEL  = CI(2) .LT. CIREF
      IF ( PINION ) THEN
        YP = YN(1)
      ELSE
        YP = YN(2)
      ENDIF
      IF ( WHEEL  ) THEN
        YW = YN(1)
      ELSE
        YW = YN(2)
      ENDIF
      WRITE ( 6, 1210 ) ROM, CI, YP, YW, CIREF
 1210 FORMAT (
     * ' * 6A * ROM      *', 2 (F11.3, 2H *),
     * ' MM * MAXIMUM ALLOWABLE TIP RADII    *' /
     * ' * 6A * CI       *', 2 (F11.3, 2H *),
     * ' MM * INVOLUTE CLEARANCE COEFFICIENT*' /
     * ' * 6A *          *', 2 (A11,   2H *),
     * '      * TIP INTERFERENCE (YES=CI<',F4.2,')*'  )
C
C     STEP 6A.3   CALCULATE THE BOTTOM CLEARANCE COEFFICIENT
C     =========
C                 CU(1)    = CLEARANCE COEFFICIENT PINION
C                 CU(2)    = CLEARANCE COEFFICIENT WHEEL
C                 CUREF = REFERENCE CLEARANCE COEFFICIENT
C
      CU(1) = ( C - RO(2) - RR(1) ) / MN
      CU(2) = ( C - RO(1) - RR(2) ) / MN
      GAPP   = CU(1) .LT. CUREF
      GAPW   = CU(2) .LT. CUREF
      IF ( GAPP ) THEN
        GP = YN(1)
      ELSE
        GP = YN(2)
      ENDIF
      IF ( GAPW  ) THEN
        GW = YN(1)
      ELSE
        GW = YN(2)
      ENDIF
      WRITE ( 6, 1310 ) CU, GP, GW, CUREF
 1310 FORMAT (
     * ' * 6A * CU       *', 2 (F11.3, 2H *),
     * ' MM * BOTTOM CLEARANCE COEFFICIENTS *' /
     * ' * 6A *          *', 2 (A11,   2H *),
     * '      * ROOT INTERFERENCE(YES=CU<',F4.2,')*'  )
C
C     STEP 6A.4   CHECK FOR SUFFICIENT TOP LAND WIDTH COEFFICIENT
```

```
C      =========
C
C                TST(1)    = REF TRANSVERSE ARC TOOTH THICKNESS PINION
C                TST(2)    = REF TRANSVERSE ARC TOOTH THICKNESS WHEEL
C                PSIO(1)   = TIP HELIX ANGLE PINION
C                PSIO(2)   = TIP HELIX ANGLE WHEEL
C                PHIO(1)   = TIP TRANSVERSE PRESSURE ANGLE PINION
C                PHIO(2)   = TIP TRANSVERSE PRESSURE ANGLE WHEEL
C                TOT(1)    = TRANSVERSE ARC TOOTH TIP THICKNESS PINION
C                TOT(2)    = TRANSVERSE ARC TOOTH TIP THICKNESS WHEEL
C                BETAH(1)  = PROJECTED TRANSVERSE HALF TIP ANGLE PINION
C                BETAH(2)  = PROJECTED TRANSVERSE HALF TIP ANGLE WHEEL
C                TO(1)     = NORMAL CHORDAL TOOTH TIP THICKNESS PINION
C                TO(2)     = NORMAL CHORDAL TOOTH TIP THICKNESS WHEEL
C                CW(1)     = TOP LAND WIDTH COEFFICIENT PINION
C                CW(2)     = TOP LAND WIDTH COEFFICIENT WHEEL
C                CWREF = REFERENCE TOP LAND WIDTH COEFFICIENT
C
       TST(1) = ( MN * ( 0.5 * PI + 2.0 * X(1) * TAN(PHIC) )
     *         - BN(1) / COS(PHIC) ) / COS(PSIS)
       TST(2) = ( MN * ( 0.5 * PI + 2.0 * X(2) * TAN(PHIC) )
     *         - BN(2) / COS(PHIC) ) / COS(PSIS)
       PSIO(1) = ATAN ( RO(1) * TAN(PSIS) / RS(1) )
       PSIO(2) = ATAN ( RO(2) * TAN(PSIS) / RS(2) )
       PHIO(1) = ACOS( RB(1) / RO(1) )
       PHIO(2) = ACOS( RB(2) / RO(2) )
       TOT(1) = 2.0 * RO(1) * ( 0.5 * TST(1) / RS(1) + INV(PHIS)
     * - INV(PHIO(1)) )
       TOT(2) = 2.0 * RO(2) * ( 0.5 * TST(2) / RS(2) + INV(PHIS)
     * - INV(PHIO(2)) )
       BETAH(1) = 0.5 * TOT(1) * ( COS(PSIO(1)) ** 2 )  / RO(1)
       BETAH(2) = 0.5 * TOT(2) * ( COS(PSIO(2)) ** 2 )  / RO(2)
       ANI = RTD(PSIO(1)) + 0.00005
       ANJ = RTD(PSIO(2)) + 0.00005
       ANK = RTD(PHIO(1)) + 0.00005
       ANL = RTD(PHIO(2)) + 0.00005
       ANO = RTD(BETAH(1)) + 0.00005
       ANP = RTD(BETAH(2)) + 0.00005
       TO(1) = 2.0 * RO(1) * SIN(BETAH(1)) / COS(PSIO(1))
       TO(2) = 2.0 * RO(2) * SIN(BETAH(2)) / COS(PSIO(2))
       CW(1) = TO(1) / MN
       CW(2) = TO(2) / MN
       LANDP    = CW(1) .LT. CWREF
       LANDW    = CW(2) .LT. CWREF
       IF ( LANDP ) THEN
         LP = YN(1)
       ELSE
         LP = YN(2)
       ENDIF
       IF ( LANDW ) THEN
         LW = YN(1)
       ELSE
         LW = YN(2)
       ENDIF
       WRITE ( 6, 1410 ) TST,
     * NMS(ANI), NS(ANI), NR(ANI), NMS(ANJ), NS(ANJ), NR(ANJ),
     * NMS(ANK), NS(ANK), NR(ANK), NMS(ANL), NS(ANL), NR(ANL), TOT,
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```
     * NMS(ANO), NS(ANO), NR(ANO), NMS(ANP), NS(ANP), NR(ANP),
     * TO, CW, LP, LW, CWREF
 1410 FORMAT (
     * ' * 6A * TST     *', 2 (F11.3, 2H *),
     * ' MM * REF TRANSVERSE ARC TOOTH WIDTH*' /
     * ' * 6A * PSIO    *', 2 (I4,1H ,I2.2,1H',I2.2,1H",2H *),
     * ' DEG* TIP HELIX ANGLES               *'  /
     * ' * 6A * PHIO    *', 2 (I4,1H ,I2.2,1H',I2.2,1H",2H *),
     * ' DEG* TIP TRANSVERSE PRESSURE ANGLES*'  /
     * ' * 6A * TOT     *', 2 (F11.3, 2H *),
     * ' MM * TIP TRANSVERSE ARC TOOTH WIDTH*' /
     * ' * 6A * BETAH   *', 2 (I4,1H ,I2.2,1H',I2.2,1H",2H *),
     * ' DEG* HALF TIP TRANSVERSE ANGLES     *'  /
     * ' * 6A * TO      *', 2 (F11.3, 2H *),
     * ' MM * NORMAL TOP LAND WIDTHS         *' /
     * ' * 6A * CW      *', 2 (F11.3, 2H *),
     * '      * TOP LAND WIDTH COEFFICIENTS   *'  /
     * ' * 6A *         *', 2 (A11,    2H *),
     * '      * LAND TOO SMALL    (YES=CW<',F4.2,')*'   )
      IF ( IOPT .EQ. 1 ) CALL PAUS
C
C     STEP 6A.5   CALCULATE THE CHORDAL DIMENSIONS AT THE MIDPOINT OF
C     ========= THE INVOLUTE
C
C                 RMID(1)    = RADIUS TO MID POINT OF INVOLUTE OF PINION
C                 RMID(2)    = RADIUS TO MID POINT OF INVOLUTE OF WHEEL
C                 PSIR(1)    = PINION HELIX ANGLE AT INVOLUTE MIDPOINT RAD
C                 PSIR(2)    = WHEEL HELIX ANGLE AT INVOLUTE MIDPOINT RAD
C                 PHIR(1)    = PINION TRANSVERSE PRESSURE ANGLE @ I.M.R.
C                 PHIR(2)    = WHEEL TRANSVERSE PRESSURE ANGLE @ I.M.R.
C                 TRT(1)     = PINION TRANSVERSE ARC TOOTH THICKNESS
C                 TRT(2)     = WHEEL TRANSVERSE ARC TOOTH THICKNESS
C                 BETARH(1)  = PROJECTED TRANSVERSE HALF ANGLE PINION
C                 BETARH(2)  = PROJECTED TRANSVERSE HALF ANGLE WHEEL
C                 TNC(1)     = NORMAL CHORDAL TOOTH THICKNESS PINION
C                 TNC(2)     = NORMAL CHORDAL TOOTH THICKNESS WHEEL
C                 ANC(1)     = NORMAL CHORDAL TOOTH ADDENDUM PINION
C                 ANC(2)     = NORMAL CHORDAL TOOTH ADDENDUM WHEEL
C                 SPAN(1)    = SPAN DIMENSION FOR PINION
C                 SPAN(2)    = SPAN DIMENSION FOR WHEEL
C                 STEETH(1)  = NUMBER OF PINION TEETH TO BE SPANNED
C                 STEETH(2)  = NUMBER OF WHEEL TEETH TO BE SPANNED
C                 FMIN(1)    = MINIMUM FACE WIDTH FOR SPAN DIM OF PINION
C                 FMIN(2)    = MINIMUM FACE WIDTH FOR SPAN DIM OF WHEEL
C
      RMID(1) = ( RO(1) + BIGRF(1) ) / 2.0
      RMID(2) = ( RO(2) + BIGRF(2) ) / 2.0
      PSIR(1) = ATAN ( RMID(1) * TAN(PSIS) / RS(1) )
      PSIR(2) = ATAN ( RMID(2) * TAN(PSIS) / RS(2) )
      PHIR(1) = ACOS( RB(1) / RMID(1) )
      PHIR(2) = ACOS( RB(2) / RMID(2) )
      TRT(1) = 2.0 * RMID(1) * ( 0.5 * TST(1) / RS(1) + INV(PHIS)
     * - INV(PHIR(1)) )
      TRT(2) = 2.0 * RMID(2) * ( 0.5 * TST(2) / RS(2) + INV(PHIS)
     * - INV(PHIR(2)) )
      BETARH(1) = 0.5 * TRT(1) * ( COS(PSIR(1)) ** 2 )  / RMID(1)
      BETARH(2) = 0.5 * TRT(2) * ( COS(PSIR(2)) ** 2 )  / RMID(2)
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```
      TNC(1) = 2.0 * RMID(1) * SIN(BETARH(1)) / COS(PSIR(1))
      TNC(2) = 2.0 * RMID(2) * SIN(BETARH(2)) / COS(PSIR(2))
      ANC(1) = RO(1) - RMID(1) * COS(BETARH(1))
      ANC(2) = RO(2) - RMID(2) * COS(BETARH(2))
C
C     SPAN GAUGE DIMENSIONS
C     SELECT NUMBER OF TEETH TO BE GAUGED
C
C     CALCULATE PSIB TO ENABLE THE CALCULATION OF THE MINIMUM FACE WIDTH
      IF ( SPUR ) THEN
        PSIB = 0.0
      ELSE
        PSIB = ATAN( TAN(PSIS) * COS(PHIS) )
      ENDIF
      DO 1510 N=1,2
      IF ( N .EQ. 2 .AND. RTEETH .GT. 0 ) THEN
        STEETH(N) = 0
        SPAN(N)   = 0.0
      ELSE
        STEETH(N) = INT ( NT(N) / 9.0 + 1.0 + 1.75 * X(N) )
C
C       CALCULATE SPAN DIMENSION
C
 1520   SPAN(N) = MN * ( NT(N) * COS(PHIC) * INV(PHIS)
     *          + 2.0 * X(N) * SIN(PHIC)
     *          + PI * COS(PHIC) * ( STEETH(N) - 0.5) ) - BN(N)
C
C       CHECK FOR SUFFICIENT FACE WIDTH
C
        FMIN(N) = SPAN(N) * SIN(PSIB)
        IF ( F .LE. FMIN(N) + MN ) THEN
          STEETH(N) = STEETH(N) - 1
          GOTO 1520
        ENDIF
      ENDIF
 1510 CONTINUE
      WRITE ( 6, 1530 ) RMID, ANC, TNC, SPAN, STEETH
      IF ( .NOT. SPUR ) WRITE ( 6, 1540 ) FMIN
 1530 FORMAT (
     * ' * 6A * RMID    *', 2 (F11.3, 2H *),
     * ' MM * RADII TO MID POINT OF INVOLUTE*' /
     * ' * 6A * ANC     *', 2 (F11.3, 2H *),
     * ' MM * CHORDAL HEIGHTS @ RMID         *' /
     * ' * 6A * TNC     *', 2 (F11.3, 2H *),
     * ' MM * CHORDAL WIDTHS @ RMID          *' /
     * ' * 6A * SPAN    *', 2 (F11.3, 2H *),
     * ' MM * SPAN DIMENSION                 *' /
     * ' * 6A * STEETH *', 2 (F11.3, 2H *),
     * '      * OVER NUMBER OF TEETH         *'  )
 1540 FORMAT (
     * ' * 6A * FMIN    *', 2 (F11.3, 2H *),
     * ' MM * MINIMUM FACE WIDTH FOR SPAN DIM' )
C
C     STEP 6A.6   CALCULATE THE SLIDE/ROLL RATIO
C     ========
C
C                 EPSLON(1) = SLIDE/ROLL OF PINION
```

```
C                 EPSLON(2) = SLIDE/ROLL OF WHEEL
C
      EPSLON(1) = SQRT ( RO(1) * RO(1) - RB(1) * RB(1) ) * NT(2)
     *            / ( ( SIN(PHIT) * C - SQRT ( RO(1) * RO(1)
     *            - RB(1) * RB(1) ) ) * NT(1) ) - 1.0
      EPSLON(2) = SQRT ( RO(2) * RO(2) - RB(2) * RB(2) ) * NT(1)
     *            / ( ( SIN(PHIT) * C - SQRT ( RO(2) * RO(2)
     *            - RB(2) * RB(2) ) ) * NT(2) ) - 1.0
      WRITE ( 6, 1610 ) EPSLON
 1610 FORMAT (
     * ' * 6A * EPSLON *', 2 (F11.3, 2H *),
     * '     * SLIDE/ROLL RATIOS                 *'  )
      GO TO 257
C
C     STEP 6B  IF NON-CONJUGATE ACTION THEN PRINT ERRROR MESSAGE
C     =======  AND STOP RUN AFTER COMPLETION OF STEPS 7, 8 AND 9.
C
 2000 WRITE ( 6, 2010 )
      GO TO 900
 2010 FORMAT ( 1H0, /
     * 8X, 'ANALYSIS INTERUPTED DUE TO NON-CONJUGATE ACTION.      '//
     * 8X, 'MF AND / OR MP MUST BE INCREASED BY A CHANGE  '/
     * 8X, 'IN FACE WIDTH AND / OR HELIX ANGLE AND / OR    ' /
     * 8X, 'ADDENDA. ' )
C
C     STEP 6C  IF TIP INTERFERENCE THEN PRINT ERRROR MESSAGE AND
C     ======= STOP RUN AFTER COMPLETION OF STEPS 7, 8 AND 9.
C
 3000 IF ( IOPT .GT. 1 ) WRITE ( 6, 2002 )
 3001 CONTINUE
      WRITE ( 6, 3010 )
      DLTAR3(1) = RO(1) - ( SQRT ( ( C * SIN(PHIT)
     *            - SQRT ( ( BIGRF(2)+CIREF*MN) ** 2 - RB(2) ** 2 ) ) ** 2
     *            + RB(1) ** 2 ) ) + 0.0001 + DLTARO(1)
      DLTAR3(2) = RO(2) - ( SQRT ( ( C * SIN(PHIT)
     *            - SQRT ( ( BIGRF(1)+CIREF*MN) ** 2 - RB(1) ** 2 ) ) ** 2
     *            + RB(2) ** 2 ) ) + 0.0001 + DLTARO(2)
      IF ( DLTAR3(1) .LT. 0.0 ) DLTAR3(1) = 0.0
      IF ( DLTAR3(2) .LT. 0.0 ) DLTAR3(2) = 0.0
C     CHECK FOR INSUFFICIENT BOTTOM CLEARANCE
      IF ( GAPP .OR. GAPW ) GO TO 4001
C     CHECK FOR INSUFFICIENT TOP LAND WIDTH COEFFICIENT
      IF ( LANDP .OR. LANDW ) GO TO 5001
      GO TO 6000
 3010 FORMAT ( 1H0, /
     * 8X, 'ANALYSIS INTERUPTED DUE TO TIP TO ROOT INTERFERENCE.')
C
C     STEP 6D  IF INSUFFICIENT BOTTOM CLEARANCE THEN PRINT ERRROR MESSAGE
C     =======
C
 4000 IF ( IOPT .GT. 1 ) WRITE ( 6, 2002 )
 4001 WRITE ( 6, 4010 )
      DLTAR4(1) = RO(1) - C + RR(2) + CUREF * MN + 0.001 + DLTARO(1)
      DLTAR4(2) = RO(2) - C + RR(1) + CUREF * MN + 0.001 + DLTARO(2)
      IF ( DLTAR4(1) .LT. 0.0 ) DLTAR4(1) = 0.0
      IF ( DLTAR4(2) .LT. 0.0 ) DLTAR4(2) = 0.0
C     CHECK FOR INSUFFICIENT TOP LAND WIDTH COEFFICIENT
```

```
      IF ( LANDP .OR. LANDW ) GO TO 5001
      GO TO 6000
 4010 FORMAT ( 1H0, /
     * 8X, 'ANALYSIS INTERRUPTED DUE TO INSUFFICIENT BOTTOM CLEARANCE')
C
C     STEP 6E  IF INSUFFICIENT TOP LAND WIDTH THEN PRINT ERRROR MESSAGE
C     =======
C
 5000 IF ( IOPT .GT. 1 ) WRITE ( 6, 2002 )
 5001 WRITE ( 6, 5010 )
 5002 CONTINUE

      IF ( .NOT. LANDP ) GOTO 5003
C
C     PINION FIRST
C     CALCULATE NEW PHIO(1)
      C1 = 0.5 * TST(1) / RS(1) + INV(PHIS)
      C2 = 0.5 * CWREF * MN / ( RB(1) * COS(PSIO(1)) )
C
      IF ( ICODE .NE. 0 ) GO TO 5430
C
      PHIO0 = PHIO(1)
      PHIO1 = PHIO0 - ( C1 - TAN(PHIO0) + PHIO0 - C2 * COS(PHIO0) )
     *       / ( TAN(PHIO0) * ( C1 - 2.0 * TAN(PHIO0) + PHIO0) )
      PHIO2 = PHIO1 - ( C1 - TAN(PHIO1) + PHIO1 - C2 * COS(PHIO1) )
     *       / ( TAN(PHIO1) * ( C1 - 2.0 * TAN(PHIO1) + PHIO1) )
      PHIO0  = PHIO2 - ( C1 - TAN(PHIO2) + PHIO2 - C2 * COS(PHIO2) )
     *        / ( TAN(PHIO2) * ( C1 - 2.0 * TAN(PHIO2) + PHIO2) )
      PHIO(1) = PHIO0
      GO TO 5450
C
C     FULL ITERATION FOR PHIO
C
 5430 PHIO2 = PHIO(1)
 5440 PHIO1 = PHIO2
      PHIO2 = PHIO1 - ( C1 - TAN(PHIO1) + PHIO1 - C2 * COS(PHIO1) )
     *       / ( TAN(PHIO1) * ( C1 - 2.0 * TAN(PHIO1) + PHIO1) )
      IF ( ABS ( PHIO2 - PHIO1 ) .GE. 1.0E-08 ) GO TO 5440
      PHIO(1) = PHIO2
 5450 CONTINUE
      DLTAR5(1) = ABS( RO(1) - RB(1) / COS(PHIO(1)) )
     *            + DLTARO(1) + 0.001
C
      IF ( DLTAR5(1) .LT. 0.0 ) DLTAR5(1) = 0.0
C
5003  CONTINUE

      IF ( .NOT. LANDW ) GOTO 6000
C
C     WHEEL NEXT
C     CALCULATE NEW PHIO(2)
      C1 = 0.5 * TST(2) / RS(2) + INV(PHIS)
      C2 = 0.5 * CWREF * MN / ( RB(2) * COS(PSIO(2)) )
C
      IF ( ICODE .NE. 0 ) GO TO 5460
C
      PHIO0 = PHIO(2)
```

```
      PHIO1 = PHIO0 - ( C1 - TAN(PHIO0) + PHIO0 - C2 * COS(PHIO0) )
     *      / ( TAN(PHIO0) * ( C1 - 2.0 * TAN(PHIO0) + PHIO0) )
      PHIO2 = PHIO1 - ( C1 - TAN(PHIO1) + PHIO1 - C2 * COS(PHIO1) )
     *      / ( TAN(PHIO1) * ( C1 - 2.0 * TAN(PHIO1) + PHIO1) )
      PHIO0 = PHIO2 - ( C1 - TAN(PHIO2) + PHIO2 - C2 * COS(PHIO2) )
     *      / ( TAN(PHIO2) * ( C1 - 2.0 * TAN(PHIO2) + PHIO2) )
      PHIO(2) = PHIO0
      GO TO 5480
C
C     FULL ITERATION FOR PHIO
C
 5460 PHIO2 = PHIO(2)
 5470 PHIO1 = PHIO2
      PHIO2 = PHIO1 - ( C1 - TAN(PHIO1) + PHIO1 - C2 * COS(PHIO1) )
     *      / ( TAN(PHIO1) * ( C1 - 2.0 * TAN(PHIO1) + PHIO1) )
      IF ( ABS ( PHIO2 - PHIO1 ) .GE. 1.0E-08 ) GO TO 5470
      PHIO(2) = PHIO2
 5480 CONTINUE
      DLTAR5(2) = ABS( RO(2) - RB(2) / COS(PHIO(2)) )
     *          + DLTARO(2) + 0.001
C
      IF ( DLTAR5(2) .LT. 0.0 ) DLTAR5(2) = 0.0
      GO TO 6000
C
 5010 FORMAT ( 1H0, /
     * 8X, 'ANALYSIS INTERRUPTED DUE TO INSUFFICIENT TOP LAND WIDTH.')
C
C     STEP 6F  CALCULATE AND PRINT NEW VALUES THEN
C     =======  STOP RUN
C
 6000 CONTINUE
      CNT1 = 0
      CNT2 = 0
 6005 CONTINUE
      DLTARO(1) = MAX(DLTARO(1),DLTAR3(1), DLTAR4(1), DLTAR5(1) )
      DLTARO(2) = MAX(DLTARO(2),DLTAR3(2), DLTAR4(2), DLTAR5(2) )
      RO(1) = MN * ( ( 0.5 * NT(1) / COS(PSIS) ) + X(1) )
     *      + HB(1) - DLTARO(1)
      RO(2) = MN * ( ( 0.5 * NT(2) / COS(PSIS) ) + X(2) )
     *      + HB(2) - DLTARO(2)
      ZB = SQRT( RO(2)**2 - RB(2)**2 ) - SQRT( R(2)**2 - RB(2)**2 )
      ZA = SQRT( RO(1)**2 - RB(1)**2 ) - SQRT( R(1)**2 - RB(1)**2 )
      Z=ZB+ZA
      MP = Z * COS(PSIS) / ( PI * MN * COS(PHIS) )
      MT = MF + MP
      PSIO(1) = ATAN ( RO(1) * TAN(PSIS) / RS(1) )
      PSIO(2) = ATAN ( RO(2) * TAN(PSIS) / RS(2) )
      PHIO(1) = ACOS( RB(1) / RO(1) )
      PHIO(2) = ACOS( RB(2) / RO(2) )
      TOT(1) = 2.0 * RO(1) * ( 0.5 * TST(1) / RS(1) + INV(PHIS)
     * - INV(PHIO(1)) )
      TOT(2) = 2.0 * RO(2) * ( 0.5 * TST(2) / RS(2) + INV(PHIS)
     * - INV(PHIO(2)) )
      BETAH(1) = 0.5 * TOT(1) * ( COS(PSIO(1)) ** 2 )  / RO(1)
      BETAH(2) = 0.5 * TOT(2) * ( COS(PSIO(2)) ** 2 )  / RO(2)
      TO(1) = 2.0 * RO(1) * SIN(BETAH(1)) / COS(PSIO(1))
      TO(2) = 2.0 * RO(2) * SIN(BETAH(2)) / COS(PSIO(2))
```

```
      CW(1) = TO(1) / MN
      CW(2) = TO(2) / MN
      IF ( LANDP .AND. CWREF - CW(1) .GT. 0.0005 ) THEN
        DLTAR5(1) = DLTAR5(1) + ( ( CWREF - CW(1) ) * 10.0 )
        CNT1 = CNT1 + 1
        IF ( CNT1 .LT. 20 ) GOTO 6005
      ENDIF
      IF ( LANDW .AND. CWREF - CW(2) .GT. 0.0005 ) THEN
        DLTAR5(2) = DLTAR5(2) + ( ( CWREF - CW(2) ) * 10.0 )
        CNT2 = CNT2 + 1
        IF ( CNT2 .LT. 20 ) GOTO 6005
      ENDIF
      CW(1) = CW(1) - 0.0005
      CW(2) = CW(2) - 0.0005
      CU(1) = ( C - RO(2) - RR(1) ) / MN - 0.0005
      CU(2) = ( C - RO(1) - RR(2) ) / MN - 0.0005
      CI(1) = ( SQRT ( ( C * SIN(PHIT)
     *        - SQRT ( RO(1) ** 2 - RB(1) ** 2 ) ) ** 2
     *      + RB(2) ** 2 ) - BIGRF(2) ) / MN - 0.0005
      CI(2) = ( SQRT ( ( C * SIN(PHIT)
     *        - SQRT ( RO(2) ** 2 - RB(2) ** 2 ) ) ** 2
     *      + RB(1) ** 2 ) - BIGRF(1) ) / MN - 0.0005
      MT = MT - 0.0005
      IF ( DLTARO(1) .GT. 0.0 .AND. DLTARO(1) .NE. TDLTAR(1) )
     * DLTARO(1) = DLTARO(1) + 0.0005
      IF ( DLTARO(2) .GT. 0.0 .AND. DLTARO(2) .NE. TDLTAR(2) )
     * DLTARO(2) = DLTARO(2) + 0.0005
      WRITE ( 6, 829 ) OUT
      WRITE ( 6, 6001 ) DLTAR3, DLTAR4, DLTAR5
      WRITE ( 6, 6010 ) DLTARO, MF, MP, MT, CI, CU, CW
      OLDDRO(1) = DLTARO(1)
      OLDDRO(2) = DLTARO(2)
      IF ( IOPT .EQ. 1 ) CALL PAUS
      WRITE ( 6, 6020 )
      GO TO 900
 6001 FORMAT (
     * ' * 6B * DLTAR3 *', 2 (F11.3, 2H *),
     * '     * TRUNCATION APPLIED FOR CI      *'  /
     * ' * 6C * DLTAR4 *', 2 (F11.3, 2H *),
     * '     * TRUNCATION APPLIED FOR CU      *'  /
     * ' * 6D * DLTAR5 *', 2 (F11.3, 2H *),
     * '     * TRUNCATION APPLIED FOR LAND    *'  /
     * 1H , 78(1H*) )
 6010 FORMAT (
     * 8X,'TO RECTIFY RADIAL TOOTH TRUNCATION IS NECESSARY.' /
     * 8X, 'MINIMUM RADIAL TOOTH TRUNCATION OF PINION DLTARO(1) = ',
     * F11.3, ' MM' /
     * 8X, 'MINIMUM RADIAL TOOTH TRUNCATION OF WHEEL  DLTARO(2) = ',
     * F11.3, ' MM' /
     * 8X, 'THESE MODIFICATIONS YIELD THE FOLLOWING RELATIONSHIPS '/
     * 8X, 'FACE CONTACT RATIO                       MF    = ', F11.3, /
     * 8X, 'TRANSVERSE CONTACT RATIO                 MP    = ', F11.3, /
     * 8X, 'TOTAL CONTACT RATIO                      MT    = ', F11.3 /
     * 8X, 'PINION INVOLUTE CLEARANCE COEFFICIENT    CI(1) = ', F11.3 /
     * 8X, 'WHEEL  INVOLUTE CLEARANCE COEFFICIENT    CI(2) = ', F11.3 /
     * 8X, 'PINION BOTTOM CLEARANCE COEFFICIENT      CU(1) = ', F11.3 /
     * 8X, 'WHEEL BOTTOM CLEARANCE COEFFICIENT       CU(2) = ', F11.3 /
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     * 8X, 'PINION TIP WIDTH COEFFICIENT              CW(1) = ', F11.3 /
     * 8X, 'WHEEL TIP WIDTH COEFFICIENT               CW(2) = ', F11.3 / )
6020  FORMAT (
     * 8X, 'CAVEAT:' //
     * 8X, 'FOR CONJUGATE ACTION MF + MP MUST BE GREATER THAN UNITY'/
     * 8X, 'HOWEVER, WHEN THE FACE CONTACT RATIO IS SMALL, TRANSVERSE'/
     * 8X, 'CONTACT RATIOS LESS THAN UNITY ARE UNDESIRABLE.' //
     * 8X, 'REF:-  H. MERRITT  -  GEAR ENGINEERING 1971 ED. SECT 12.20')
C
C     STEP 7      MF      = FACE CONTACT RATIO
C     =======
C
  257 MF = F * SIN(PSIS) / ( PI * MN )
C     SPUR GEARS
      SPUR  = MF .EQ. 0.0
C     CONVENTIONAL HELICAL GEARS
      HELIC = MF .GT. 1.0 .AND..NOT. SPUR
C     LOW CONTACT RATIO HELICAL GEARS
      LCR   = MF .LE. 1.0 .AND..NOT. SPUR
C
C     STEP 8      ZB      = LENGTH OF APPROACH PATH
C     =======     ZA      = LENGTH OF RECSSS PATH
C
      ZB = SQRT( RO(2)**2 - RB(2)**2 ) - SQRT( R(2)**2 - RB(2)**2 )
      ZA = SQRT( RO(1)**2 - RB(1)**2 ) - SQRT( R(1)**2 - RB(1)**2 )
C
C     STEP 9      Z      = LENGTH OF LINE OF ACTION IN TRANSVERSE PLANE
C     =======     MP     = TRANSVERSE CONTACT RATIO
C                 MT     = TOTAL CONTACT RATIO
C                 CTREF  = REFERENCE TOTAL CONTACT RATIO
C
      Z=ZB+ZA
      MP = Z * COS(PSIS) / ( PI * MN * COS(PHIS) )
      MT = MF + MP
      NONCON = MT .LE. CTREF
      IF ( NONCON ) THEN
        MTYES = YN(1)
      ELSE
        MTYES = YN(2)
      ENDIF
       WRITE ( 6, 259 ) MF, MF, MP, MP, MT, MT, MTYES, MTYES
     *,CTREF
  259 FORMAT (
     * ' * 6A * MF      *', 2 (F11.3, 2H * ),
     * '     * FACE CONTACT RATIO             *'  /
     * ' * 6A * MP      *', 2 (F11.3, 2H * ),
     * '     * TRANSVERSE CONTACT RATIO       *'  /
     * ' * 6A * MT      *', 2 (F11.3, 2H * ),
     * '     * TOTAL CONTACT RATIO            *'  /
     * ' * 6A *         *', 2 (A11,   2H *),
     * '     * NON-CONJUGATE ACTION(MT<=',F4.2,')*' /
     * 1H , 78(1H*) )
      IF ( IOPT .EQ. 1 ) CALL PAUS
C     CHECK FOR NON-CONJUGATE ACTION
      IF ( NONCON ) GO TO 2000
C     CHECK FOR TIP INTERFERENCE
      IF ( PINION .OR. WHEEL ) GO TO 3000
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C      CHECK FOR INSUFFICIENT BOTTOM CLEARANCE
       IF ( GAPP .OR. GAPW ) GO TO 4000
C      CHECK FOR INSUFFICIENT TOP LAND WIDTH COEFFICIENT
       IF ( LANDP .OR. LANDW ) GO TO 5000
C      WRITE OUT STEP 1 TO STEP 9
       IF ( IOPT .GT. 1 ) WRITE ( 6, 2002 )
  260 CONTINUE
       WRITE ( 6, 832 ) NN, TYPE(ICODE+1), TITLE
  832 FORMAT ( 9H GEAR SET, I3, 5X, A14, / 1X, A68 )
       WRITE ( 6, 829 ) OUT
       WRITE ( 6, 201 ) NMS(ANG), NS(ANG), NR(ANG),
     *                  NMS(ANG), NS(ANG), NR(ANG)
  201 FORMAT (
     * ' *  1 * PHIS    *',2(I4,1H ,I2.2,1H',I2.2,1H",2H *),
     * ' DEG* TRANSVERSE PRESSURE ANGLE      *'  )
       WRITE ( 6, 202 ) NMS(ANH), NS(ANH), NR(ANH),
     *                  NMS(ANH), NS(ANH), NR(ANH)
  202 FORMAT (' *  2 * PHIT    *',2(I4,1H ,I2.2,1H',I2.2,1H",2H *),
     * ' DEG* OPERATING TRANSVERSE PRESS ANG*'  )
       WRITE ( 6, 203 ) C, C
  203 FORMAT ( ' *  3 * C       *', 2 (F11.3, 2H *),
     *          ' MM * OPERATING CENTRE DISTANCE     *' )
       WRITE ( 6, 204 ) RB
  204 FORMAT ( ' *  4 * RB      *', 2 (F11.3, 2H *),
     *          ' MM * BASE RADII                    *' )
       WRITE ( 6, 205 ) R
  205 FORMAT ( ' *  5 * R       *', 2 (F11.3, 2H *),
     *          ' MM * OPERATING PITCH RADII         *' )
       WRITE ( 6, 206 ) RO
  206 FORMAT ( ' *  6 * RO      *', 2 (F11.3, 2H *),
     *          ' MM * TIP RADII                     *' )
       WRITE ( 6, 207 ) MF, MF
  207 FORMAT ( ' *  7 * MF      *', 2 (F11.3, 2H *),
     *          '    * FACE CONTACT RATIO            *' )
       WRITE ( 6, 208 ) ZB, ZB, ZA, ZA
  208 FORMAT ( ' *  8 * ZB      *', 2 (F11.3, 2H *),
     *          ' MM * LENGTH OF APPROACH PATH       *' /
     *          ' *  8 * ZA      *', 2 (F11.3, 2H *),
     *          ' MM * LENGTH OF RECESS PATH         *' )
       WRITE ( 6, 209 ) Z, Z
  209 FORMAT ( ' *  9 * Z       *', 2 (F11.3, 2H *),
     *          ' MM * LENGTH OF LINE OF ACTION      *' )
C      STEP 10     ZC = DISTANCE FROM PITCH POINT TO STRESS POINT
C      =======
C
       IF ( HELIC ) THEN
C        CONVENTIONAL HELICAL GEARS
         ZC =          SQRT ( R(1) ** 2 - RB(1) ** 2 )
     *       - SQRT ( 0.25 * ( ( C - RO(2) + RO(1) ) ** 2 ) - RB(1) ** 2 )
       ELSE
C        SPUR OR LOW CONTACT RATIO HELICAL GEARS
         ZC = PI * MN * COS(PHIS) / COS(PSIS) - ZA
       ENDIF
       WRITE ( 6, 210 ) ZC, ZC
  210 FORMAT ( ' * 10 * ZC      *', 2 (F11.3, 2H *),
     *          ' MM * DIST STRESS TO PITCH POINTS   *' )
C
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C      STEP 11       CX      = CONTACT HEIGHT FACTOR
C      =======
C
       CX = ( SIN(PHIT) - ZC / R(1) ) * ( SIN(PHIT) + ZC / R(2) )
     *    / ( SIN(PHIT) ** 2 )
        WRITE ( 6, 211 ) CX, CX
  211 FORMAT ( ' * 11 * CX       *', 2 (F11.3, 2H *),
     *         '       * CONTACT HEIGHT FACTOR          *'  )
C
C      STEP 12      ZCH   = DISTANCE FROM PITCH POINT TO STRESS POINT - LCR
C      =======      CXH   = CONTACT HEIGHT FACTOR - LOW CONTACT RATIO GEARS
C
       IF ( LCR ) THEN
        ZCH = SQRT ( R(1) ** 2 - RB(1) ** 2 )
     *       - SQRT ( 0.25 * ( ( C - RO(2) + RO(1) ) ** 2 ) - RB(1) ** 2 )
        CXH = ( SIN(PHIT) - ZCH / R(1) ) * ( SIN(PHIT) + ZCH / R(2) )
     *       / ( SIN(PHIT) ** 2 )
        WRITE ( 6, 212 ) ZCH, ZCH, CXH, CXH
       ENDIF
  212 FORMAT ( ' * 12 * ZCH      *', 2 (F11.3, 2H *),
     *         ' MM * LENGTH OF LINE OF ACTION LCR   *'     /
     *         ' * 12 * CXH      *', 2 (F11.3, 2H *),
     *         '       * CONTACT HEIGHT FACTOR LCR      *'  )
C
C      STEP 13      PSIB    = BASE HELIX ANGLE
C      =======
C
       IF ( SPUR ) THEN
         PSIB = 0.0
       ELSE
         PSIB = ATAN( TAN(PSIS) * COS(PHIS) )
       ENDIF
       ANG = RTD(PSIB) + 0.00005
        WRITE ( 6, 213 ) NMS(ANG), NS(ANG), NR(ANG),
     *                   NMS(ANG), NS(ANG), NR(ANG)
  213 FORMAT ( ' * 13 * PSIB    *',2(I4,1H ,I2.2,1H',I2.2,1H",2H *),
     *         ' DEG* BASE HELIX ANGLE             *'  )
C
C      STEP 14      CPSI    = HELICAL OVERLAP FACTOR
C      =======
C
       IF ( LCR ) THEN
         CPSI = SQRT ( 1.0 - MF + CXH * Z * MF ** 2
     *         / ( CX * F * SIN(PSIB) ) )
       ELSE
         CPSI = 1.0
       ENDIF
        WRITE ( 6, 214 ) CPSI, CPSI
  214 FORMAT ( ' * 14 * CPSI    *', 2 (F11.3, 2H *),
     *         '       * HELICAL OVERLAP FACTOR         *'  )
C
C      STEP 15      CC      = CURVATURE FACTOR
C      =======
C
       CC = 0.5 * NT(2) * COS(PHIT) * SIN(PHIT) / ( NT(1) + NT(2) )
        WRITE ( 6, 215 ) CC, CC
  215 FORMAT ( ' * 15 * CC      *', 2 (F11.3, 2H *),
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```
     *           '    * CURVATURE FACTOR                 *'  )
C
C     STEP 16      LMIN    = MINIMUM LENGTH OF LINES OF CONTACT
C     =======      PB      = TRANSVERSE BASE PITCH
C                  MP      = TRANSVERSE CONTACT RATIO
C                  N       = LIMITING NUMBER OF LINES OF CONTACT
C                  CO      = OVERLAP COEFFICIENT
C     ****************************************************************
C     * NOTE. THESE EQUATIONS REPLACE CLAUSE 6.2.6 OF AGMA 218.01-1982 *
C     * NOTE. THESE EQUATIONS REPLACE CLAUSE 6.2.6 OF AS2938-1987      *
C     ****************************************************************
      IF ( .NOT. HELIC ) THEN
        LMIN = F
      ELSE
        PB    = PI * MN * COS(PHIS) / COS(PSIS)
        MP    = Z / PB
        N     = INT ( MP + MF )
        CO    = N - MF
        IF ( MF .GT. N ) CO = 0.0
        LMIN = PB * ( MP * N - CO - INT(MP) * FRAC(MP)
     *        - INT(CO) * ( INT(MP) + FRAC(CO) - 1 ) ) / SIN(PSIB)
C     ***********************************************************
C     *      ALTERNATE APPROXIMATION FOR LMIN                   *
C     *      SEE AGMA  218.01   EQUATION NO. 6.27               *
C     *      SEE AS2938-1987    EQUATION NO. 6.27               *
C       LMIN = 0.95 * F * Z / ( PI * COS(PHIC) )
C     ***********************************************************
      ENDIF
       WRITE ( 6, 216 ) LMIN, LMIN
  216 FORMAT ( ' * 16 * LMIN    *', 2 (F11.3, 2H *),
     *           ' MM * MIN LENGTH OF LINES OF CONTACT*' )
C
C     STEP 17      MNN    = LOAD SHARING RATIO
C     =======
C
      MNN = F / LMIN
       WRITE ( 6, 217 ) MNN, MNN
  217 FORMAT ( ' * 17 * MNN     *', 2 (F11.3, 2H * ),
     *           '    * LOAD SHARING RATIO           *' )
C
C     STEP 18      I      = PITTING RESISTANCE GEOMETRY FACTOR I
C     =======
C
      I = CX * CPSI ** 2 * CC / MNN
       WRITE ( 6, 218 ) I, I
  218 FORMAT ( 1X,78(1H*) /
     * ' * 18 * I       *',2(F11.3,2H *),
     * '    * GEOMETRY FACTOR - PITTING     *'  /
     *        1X,78(1H*) )
      IF ( IOPT .EQ. 1 ) CALL PAUS
C
C     ***************************************************************
C     *          REPEAT STEP 19 TO STEP 32 TWICE                    *
C     *                 FIRST FOR PINION                            *
C     *                  THEN FOR WHEEL                             *
C     ***************************************************************
C
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      DO 800 N = 1 , 2
      OUTPUT = N .EQ. 2
C
C     STEP 19      PSI     = HELIX ANGLE AT OPERATING PITCH DIAMETER
C     =======
C
      IF ( SPUR ) THEN
        PSI(N) = 0.0
      ELSE
        PSI(N) = ATAN ( 2.0 * R(N) * SIN(PSIS) / ( NT(N) * MN ) )
      ENDIF
      ANN(N) = RTD(PSI(N)) + 0.00005
      IF ( OUTPUT )
     *WRITE ( 6, 219 ) NMS(ANN(1)), NS(ANN(1)), NR(ANN(1)),
     *                 NMS(ANN(2)), NS(ANN(2)), NR(ANN(2))
  219 FORMAT ( ' * 19 * PSI     *',2(I4,1H ,I2.2,1H',I2.2,1H",2H *),
     *          ' DEG* OPERATING PITCH DIA HELIX ANG *'  )
C
C     STEP 20      PHIN = OPERATING NORMAL PRESSURE ANGLE
C     =======
C
      IF ( SPUR ) THEN
        PHIN(N) = PHIT
      ELSE
        PHIN(N) = ATAN ( TAN(PHIT) * COS(PSI(N)) )
      ENDIF
      ANM(N) = RTD(PHIN(N)) + 0.00005
      IF ( OUTPUT )
     *WRITE ( 6, 220 ) NMS(ANM(1)), NS(ANM(1)), NR(ANM(1)),
     *                 NMS(ANM(2)), NS(ANM(2)), NR(ANM(2))
  220 FORMAT ( ' * 20 * PHIN    *',2(I4,1H ,I2.2,1H',I2.2,1H",2H *),
     *          ' DEG* OPERATING NORMAL PRESSURE ANG *'  )
C
C     STEP 21      NE     = VIRTUAL NUMBER OF SPUR TEETH
C     =======      ROE    = VIRTUAL SPUR TIP RADIUS
C
      NE(1)  = NT(1) / ( COS(PSI(1)) ** 3 )
      NE(2)  = NE(1) * NT(2) / NT(1)
      ROE(1) = MN * ( 0.5 * NE(1) + X(1) ) + HB(1) - DLTARO(1)
      ROE(2) = MN * ( 0.5 * NE(2) + X(2) ) + HB(2) - DLTARO(2)
      IF ( OUTPUT ) WRITE ( 6, 221 ) NE, ROE
  221 FORMAT ( ' * 21 * NE      *', 2 (F11.3, 2H *),
     *          '     * VIRTUAL NUMBER OF SPUR TEETH  *'  /
     *          ' * 21 * ROE     *', 2 (F11.3, 2H *),
     *          ' MM * VIRTUAL SPUR TIP RADII         *' )
C
C     STEP 22      BETA  = ANGLE BETWEEN VIRTUAL BASE RADIUS & TIP RADIUS
C     =======      PHILN = LOAD ANGLE AT TIP
C                  PHIL  = LOAD ANGLE AT TIP OR HPSTC
C                  RBE   = EQUIVALENT BASE RADIUS
C                  RE    = EQUIVALENT OPERATING PITCH RADIUS
C                  ZNE   = EQUIVALENT LENGTH OF LINE OF ACTION
C           NB: "HPSTC" = HIGHEST POINT OF SINGLE TOOTH CONTACT
C
      BETA(N)    = ACOS ( 0.5 * NE(N) * MN * COS(PHIC) /  ROE(N) )
      PHILN(N) = TAN ( BETA(N) )
     * - ( ( 0.5 * PI + 2 * X(N) * TAN(PHIC) - BN(N) / MN) / NE(N) )
```

```
     * - INV(PHIC)
      IF ( HELIC ) THEN
C        CALCULATION OF LOAD ANGLE AT TIP FOR CONVENTIONAL HELICAL GEARS
         PHIL(N) = PHILN(N)
      ELSE
        IF ( SPUR ) THEN
          IF ( ROUGH .NE. 1 ) THEN
C           CALCULATION OF LOAD ANGLE AT HPSTC FOR ACCURATE SPUR GEARS
            PHIL(N) = PHILN(N)- 2 * ( Z / MN - PI * COS(PHIC) ) /
     *                ( NT(N) * COS(PHIC) )
          ELSE
C           CALCULATION OF LOAD ANGLE AT TIP FOR INACCURATE SPUR GEARS
            PHIL(N) = PHILN(N)
          ENDIF
        ELSE
C         CALCULATION OF LOAD ANGLE AT HPSTC FOR LCR HELICAL GEARS
          RBE(1) = 0.5 * NE(1) * MN * COS(PHIC)
          RBE(2) = RBE(1) * NT(2) / NT(1)
          RE(1)  = R(1) / ( COS(PSI(1)) ** 2 )
          RE(2)  = RE(1) * NT(2) / NT(1)
          ZNE = SQRT( ROE(2)**2-RBE(2)**2 ) - SQRT( RE(2)**2-RBE(2)**2 )
     *        + SQRT( ROE(1)**2-RBE(1)**2 ) - SQRT( RE(1)**2-RBE(1)**2 )
          PHIL(N) = PHILN(N) - 2.0 * ( ZNE / MN - PI * COS(PHIC) )
     *        / ( NE(N) * COS(PHIC) )
        ENDIF
      ENDIF
      IF ( OUTPUT ) WRITE ( 6, 222 ) BETA
  222 FORMAT ( ' * 22 * BETA    *', 2 (F11.3, 2H *),
     *         ' RAD* BASE & TIP RADII INCLUDED ANGS*' )
      IF ( OUTPUT .AND. SPUR .AND. ROUGH .EQ. 0 )
     * WRITE ( 6, 272 ) PHILN
      IF ( OUTPUT .AND. LCR ) WRITE ( 6, 272 ) PHILN
  272 FORMAT ( ' * 22 * PHILN   *', 2 (F11.3, 2H *),
     *         ' RAD* LOAD ANGLE AT TIP              *' )
          IF ( OUTPUT .AND. LCR )
     *    WRITE ( 6, 322 ) RBE, RE, ZNE, ZNE
  322 FORMAT ( ' * 22 * RBE     *', 2 (F11.3, 2H *),
     *         ' MM * EQUIVALENT BASE RADII          *' /
     *         ' * 22 * RE      *', 2 (F11.3, 2H *),
     *         ' MM * EQUIVALENT OPERATING RADII     *' /
     *         ' * 22 * ZNE     *', 2 (F11.3, 2H *),
     *         ' MM * EQUIVALENT Z FOR LCR HELICAL   *' )
      IF ( OUTPUT .AND. HELIC) WRITE ( 6, 372 ) PHIL
  372 FORMAT ( ' * 22 * PHIL    *', 2 (F11.3, 2H *),
     *         ' RAD* LOAD ANGLES AT TIP             *' )
      IF ( OUTPUT .AND. .NOT. HELIC ) THEN
        IF ( ROUGH .NE. 1 ) THEN
          WRITE ( 6, 422 ) PHIL
        ELSE
          WRITE ( 6, 372 ) PHIL
        ENDIF
      ENDIF
  422 FORMAT ( ' * 22 * PHIL    *', 2 (F11.3, 2H *),
     *         ' RAD* LOAD ANGLES AT HPSTC           *' )
C
C     STEP 23      LAMBDI = INITIAL VALUE FOR LAMBDA
C     =======
```

```
C
      IF ( HELIC .OR. ROUGH .EQ. 1 ) THEN
        LAMBDA(N) = 0.75 * PHIC
      ELSE
        LAMBDA(N) = PHIC
      ENDIF
      LAMBAI(N) = LAMBDA(N)
      IF ( OUTPUT ) WRITE ( 6, 223 ) LAMBAI
  223 FORMAT ( ' * 23 * LAMBAI *', 2 (F11.3, 2H *),
     *          ' RAD* LAMBDA - INITIAL VALUE          *' )
C
C     STEP 24      K1     = CONSTANT
C     =======      K2     = CONSTANT
C                  K3     = CONSTANT
C     **************************************************************
C     *    STEP 23 TO STEP 27 CONSTITUTE THE LAMBDA METHOD FOR THE   *
C     *     DETERMINATION OF THE ACTUAL PARABOLA HEIGHT AND WIDTH    *
C     **************************************************************
      K1(N) = ( HA(N) - RT(N) ) / MN - X(N)
     *        + 0.5 * BN(N) / ( MN * TAN(PHIC) )
      IF ( K1(N) .EQ. 0.0 ) K1(N) = 1.0E-16
      K2(N) = 0.25 * PI + ( HA(N) * TAN(PHIC) + RT(N)
     *        * ( 1.0 - SIN(PHIC) ) / COS(PHIC)) / MN
      K2(N) = K2(N) -  ( ( DELTAO(N) / COS(PHIC) ) / MN )
      K3(N) = 0.5 * NE(N) / K1(N)
      IF ( OUTPUT ) WRITE ( 6, 224 ) K1, K2, K3
  224 FORMAT ( ' * 24 * K1     *', 2 (F11.3, 2H *),
     *          ' MM * CONSTANT - K1                   *' /
     *          ' * 24 * K2     *', 2 (F11.3, 2H *),
     *          ' MM * CONSTANT - K2                   *' /
     *          ' * 24 * K3     *', 2 (F11.3, 2H *),
     *          '    * CONSTANT - K3                   *'  )
C
C     STEP 25      K4       = VARIABLE
C     =======      K5       = VARIABLE
C                  K6       = VARIABLE
C                  K7       = VARIABLE
C                  K8       = VARIABLE
C                  K9       = VARIABLE
C
C                  KOUNT1   = NUMBER OF ITERATIONS OF LAMBDA(N) PREFORMED
C                  KOUNT2   = NUMBER OF ITERATIONS OF K6(N) PREFORMED
C
      KOUNT1 = 0
   25 KOUNT1 = KOUNT1 + 1
C
C     STOP ANALYSIS IF MORE THAN 20 ITERATIONS OF LABMDA(N)
C
      IF ( KOUNT1 .GT. 20 )  GO TO 325
C
      K4(N) = 0.5 * PI - LAMBDA(N) - 2.0 * K2(N) / NE(N)
      K5(N) = K3(N) / ( K3(N) + 1.0 ) * K4(N)
      K6(N) = K5(N) - ( TAN(K5(N)) + K5(N) * K3(N) - K4(N) * K3(N) )
     *        / ( ( 1.0 / COS(K5(N)) ) ** 2 + K3(N) )
C
      KOUNT2 = 0
   75 KOUNT2 = KOUNT2 + 1
```

```
C
C     STOP ANALYSIS IF MORE THAN 20 ITERATIONS OF K6(N)
C
      IF ( KOUNT2 .GT. 20 ) GO TO 425
C
      K7(N) = K6(N) - ( TAN(K6(N)) + K6(N) * K3(N) - K4(N) * K3(N) )
     *        / ( ( 1.0 / COS(K6(N)) ) ** 2 + K3(N) )
C     ***************************************************************
C     *              ITERATIVE LOOP FOR MORE EXACT ANSWERS          *
C     *                    ACTIVATED BY ICODE = 1                   *
C     ***************************************************************
      IF ( ICODE .EQ. 1 .AND.
     *   ABS ( TAN(K7(N)) / K3(N) + K7(N) - K4(N) ) .GT. 1.0E-08 ) THEN
        K6(N) = K7(N)
        GO TO 75
      ENDIF
C     ***************************************************************
      K8(N) = 0.5 * PI - LAMBDA(N) - K7(N)
      K9(N) = RT(N) / MN + K1(N) / COS(K7(N))
C
C     STEP 26      TE      = INSCRIBED PARABOLA WIDTH
C     =======      HE      = INSCRIBED PARABOLA HEIGHT
C
      TE(N) = MN * ( NE(N) * SIN(K8(N)) - 2.0 * K9(N) * COS(LAMBDA(N)) )
      HE(N) = MN * ( 0.5 * NE(N) * ( COS(PHIC)
     * / COS(PHIL(N)) - COS(K8(N)) ) + K9(N) * SIN(LAMBDA(N)) )
C
C     STEP 27      LAMBDA = FINAL VALUE FOR LAMBDA
C     =======
C
      LAMBA1(N) = ATAN ( 0.25 * TE(N) / HE(N) )
C     ***************************************************************
C     *              ITERATIVE LOOP FOR MORE EXACT ANSWERS          *
C     *                    ACTIVATED BY ICODE = 1                   *
C     ***************************************************************
      IF ( ICODE .EQ.1 .AND. ABS(LAMBA1(N)-LAMBDA(N)) .GT. 1.0E-08 )THEN
        LAMBDA(N) = ABS( LAMBA1(N) )
        GO TO 25
      ENDIF
C     ***************************************************************
      IF ( ICODE .EQ.0 .AND. ABS(LAMBA1(N)-LAMBDA(N)) .GT. 0.05 ) THEN
        LAMBDA(N) = ABS( LAMBA1(N) )
        GO TO 25
      ENDIF
      LAMBDA(N) = LAMBA1(N)
      GO TO 27
C
C     IF TOO MANY ITERATIONS OF LAMBDA(N) WRITE MESSAGE & STOP ANALYSIS
C
  325 CONTINUE
      WRITE ( 6, 375 )
      GO TO 900
  375 FORMAT ( 1X, 78 ( 1H* )/ ' *  ANALYSIS STOPPED DUE TO NON ',
     *     'CONVERGENCE OF LAMBDA  CHECK INPUT DATA        *' /
     *          1X , 78 ( 1H* ) )
C
C     IF TOO MANY ITERATIONS OF K6(N) WRITE MESSAGE & STOP ANALYSIS
```

```
C
  425 CONTINUE
      WRITE ( 6, 475 )
      GO TO 900
  475 FORMAT ( 1X, 78 ( 1H* )/ ' *  ANALYSIS STOPPED DUE TO NON ',
     *     'CONVERGENCE OF K6  CHECK INPUT DATA          *' /
     *          1X , 78 ( 1H* ) )
C
C     CONTINUE ANALYSIS
C
   27 CONTINUE
      IF ( OUTPUT ) WRITE ( 6, 225 ) K4, K5, K6
      IF ( OUTPUT ) WRITE ( 6, 275 ) K7, K8, K9
  225 FORMAT ( ' * 25 * K4      *', 2 (F11.3, 2H *),
     *         '     * VARIABLE - K4                   *' /
     *         ' * 25 * K5      *', 2 (F11.3, 2H *),
     *         ' RAD* VARIABLE - K5                   *' /
     *         ' * 25 * K6      *', 2 (F11.3, 2H *),
     *         ' RAD* VARIABLE - K6                   *' )
  275 FORMAT ( ' * 25 * K7      *', 2 (F11.3, 2H *),
     *         ' RAD* VARIABLE - K7                   *' /
     *         ' * 25 * K8      *', 2 (F11.3, 2H *),
     *         ' RAD* VARIABLE - K8                   *' /
     *         ' * 25 * K9      *', 2 (F11.3, 2H *),
     *         ' MM * VARIABLE - K9                   *' )
      IF ( OUTPUT ) WRITE ( 6, 226 ) TE, HE
  226 FORMAT ( ' * 26 * TE      *', 2 (F11.3, 2H *),
     *         ' MM * INSCRIBED PARABOLA - WIDTHS    *' /
     *         ' * 26 * HE      *', 2 (F11.3, 2H *),
     *         ' MM * INSCRIBED PARABOLA - HEIGHTS   *' )
      IF ( OUTPUT ) WRITE ( 6, 227 ) LAMBDA
  227 FORMAT ( ' * 27 * LAMBDA *', 2 (F11.3, 2H *),
     *         ' RAD* LAMBDA - FINAL VALUES           *' )
      IF ( OUTPUT .AND. IOPT .EQ. 1 ) CALL PAUS
C
C     STEP 28     OMEGA = LOAD INCLINATION ANGLE IN DEGREES / 100.0
C     =======     CH    = HELICAL FACTOR
C
      IF ( HELIC ) THEN
        OMEGA = ATAN( TAN(PSI(N)) * SIN(PHIN(N)) ) * 1.8 / PI
        CH(N) = 1.0 / ( 1.0 - SQRT( OMEGA - OMEGA ** 2 ) )
      ELSE
        CH(N) = 1.0
      ENDIF
      IF ( BUTT(N) .EQ. 1 ) CH(N) = CH(N) * 1.1
      IF ( OUTPUT ) WRITE ( 6, 228 ) CH
  228 FORMAT ( ' * 28 * CH      *', 2 (F11.3, 2H *),
     *         '     * HELICAL FACTOR                  *'  )
C
C     STEP 29     KPSI   = HELIX ANGLE FACTOR
C     =======
C
      IF ( HELIC ) THEN
        KPSI(N) = COS(PSI(N)) * COS(PSIS)
      ELSE
        KPSI(N) =  1.0
      ENDIF
```

```
      IF ( OUTPUT ) WRITE ( 6, 229 ) KPSI
  229 FORMAT ( ' * 29 * KPSI      *', 2 (F11.3, 2H *),
     *         '      * HELIX ANGLE FACTOR              *'  )
C
C     STEP 30     Y       = TOOTH FORM FACTOR
C     =======
C
      Y(N) = KPSI(N) * TE(N) ** 2 * COS(PHIN(N)) / ( MN * COS(PHIL(N))
     *     * ( 6.0 * HE(N) / CH(N) - TE(N) * TAN(PHIL(N)) ) )
      IF ( OUTPUT ) WRITE ( 6, 230 ) Y
  230 FORMAT ( ' * 30 * Y         *', 2 (F11.3, 2H *),
     *         '      * TOOTH FORM FACTORS              *'  )
C
C     STEP 31     B       = OPERATING DEDENDUM
C     =======     RF      = MINIMUM ROOT FILLET RADIUS
C                 H       = DOLAN - BROGHAMER FACTOR - H
C                 L       = DOLAN - BROGHAMER FACTOR - L
C                 M       = DOLAN - BROGHAMER FACTOR - M
C                 KF      = STRESS CONCENTRATION FACTOR
C
      B(N) = R(N) + HA(N) - MN * ( X(N) + 0.5 * NT(N) / COS(PSIS)
     *     - 0.5 * BN(N) / ( MN * SIN(PHIC) ) )
      RF(N) = ( B(N)-RT(N)) ** 2 / ( 0.5 * MN * NE(N)+B(N)-RT(N))+RT(N)
      H(N) = 0.34 - 0.45836624 * PHIN(N)
      L(N) = H(N) - 0.03
      M(N) = 0.25 + 0.5729578 * PHIN(N)
      KF(N) = H(N) + ( ( TE(N) / RF(N) ) ** L(N) )
     * * ( ( TE(N) /HE(N) ) ** M(N) )
      IF ( OUTPUT ) WRITE ( 6, 231 ) B, RF, H
      IF ( OUTPUT ) WRITE ( 6, 281 ) L, M, KF
  231 FORMAT ( ' * 31 * B         *', 2 (F11.3, 2H *),
     *         ' MM * OPERATING DEDENDA              *' /
     *         ' * 31 * RF        *', 2 (F11.3, 2H *),
     *         ' MM * MINIMUM ROOT FILLET RADII      *' /
     *         ' * 31 * H         *', 2 (F11.3, 2H *),
     *         '      * DOLAN - BROGHAMER FACTOR - H  *'  )
  281 FORMAT ( ' * 31 * L         *', 2 (F11.3, 2H *),
     *         '      * DOLAN - BROGHAMER FACTOR - L  *'  /
     *         ' * 31 * M         *', 2 (F11.3, 2H *),
     *         '      * DOLAN - BROGHAMER FACTOR - M  *'  /
     *         ' * 31 * KF        *', 2 (F11.3, 2H *),
     *         '      * STRESS CONCENTRATION FACTORS  *'  )
C
C     STEP 32     J       = STRENGTH GEOMETRY FACTOR
C     =======
C
      J(N) = Y(N) * CPSI / ( KF(N) * MNN )
      IF ( OUTPUT ) WRITE ( 6, 232 ) J
  232 FORMAT (1X,78(1H*)/
     * ' * 32 * J         *',2(F11.3,2H *),
     * '      * GEOMETRY FACTORS - STRENGTH       *'  /
     * 1X, 78(1H*) )
C
  800 CONTINUE
C
      IF ( IOPT .EQ. 1 ) THEN
        CALL PAUS
```

```
      ELSE
        WRITE ( 6, 2002 )
      ENDIF
C
C     PRINT OUT GEAR SUMMARY
C
      IF ( RTEETH .EQ. 0 ) THEN
        CRACK = 0.0
      ELSE
        CRACK = C - CNTRS
        ANM(1) = RTD(PHIC) + 0.00005
        ANM(2) = RTD(PHIC) + 0.00005
      ENDIF
      RMID(2) = RMID(2) - CRACK
      HAB(1) = HA(1) + HB(1)
      HAB(2) = HA(2) + HB(2)
      A(1) = RO(1) - R(1)
      A(2) = RO(2) - R(2)
      AB(1) = A(1) + B(1)
      AB(2) = A(2) + B(2)
      D0(1) = 2.0 * RO(1)
      D0(2) = 2.0 * ( RO(2) - CRACK )
      D(1) = 2.0 * R(1)
      D(2) = 2.0 * ( R(2) - CRACK )
      DR(1) = 2.0 * RR(1)
      DR(2) = 2.0 * ( RR(2) - CRACK )
      DS(1) = 2.0 * RS(1)
      DS(2) = 2.0 * ( RS(2) - CRACK )
      CM(1) = CU(1) * MN
      CM(2) = CU(2) * MN
      CN(1) = CI(1) * MN
      CN(2) = CI(2) * MN
      ITEETH(1) = STEETH(1)
      ITEETH(2) = STEETH(2)
      NWT = NW
      IF ( RTEETH .GT. 0 ) NWT = RTEETH
      CT = C
      IF ( RTEETH .GT. 0 ) CT = CNTRS
      WRITE ( 6, 10010 ) NN, TYPE(ICODE+1), TITLE, OTT, OSS
      WRITE ( 6, 10020 ) MN, MN,
     *                   NX(AN1), NY(AN1), NZ(AN1),
     *                   NX(AN1), NY(AN1), NZ(AN1),
     *                   RT, HA, HB, DELTAO
      IF ( IOPT .EQ. 1 ) CALL PAUS
      WRITE ( 6, 10030 ) NP, NWT
      WRITE ( 6, 10031 ) NX(AN2), NY(AN2), NZ(AN2),
     *                   NX(AN2), NY(AN2), NZ(AN2)
      WRITE ( 6, 10032 ) X, DLTARO,
     *                   NMS(ANM(1)), NS(ANM(1)), NR(ANM(1)),
     *                   NMS(ANM(2)), NS(ANM(2)), NR(ANM(2)),
     *                   DS, D, CT, CT, A, B, AB, HAB,
     *                   CN, CM, EPSLON, MT, MT
      WRITE ( 6, 10040 ) I, I, J
      IF ( IOPT .EQ. 1 ) CALL PAUS
      WRITE ( 6, 10050 ) QV, BN, D0,DR,RMID, ANC, TNC, SPAN, ITEETH, TO
      IF ( IOPT .NE. 1 ) THEN
        WRITE ( 6, 10060 )
```

```
      ENDIF
10010 FORMAT ( 9H GEAR SET, I3, 5X, A14 / 1X, A68 /
     *' THE FOLLOWING TABULATION SHOULD BE INCLUDED ON ENGINEERING',
     *' DRAWINGS.' //
     *'    ', 73(1H*) /
     *'    *', 5X, A42, A19 , 5X, '*' /
     *'    ', 73(1H*) )
10020 FORMAT (
     *'    * HOB OR COUNTERPART RACK DETAILS    *UNIT *     PINION    *',
     *'      WHEEL    *' / 3H    , 73(1H*) /
     *'    * NORMAL MODULE                      * MM  *', 2(F12.3, 3H   *)/
     *'    * NORMAL PRESSURE ANGLE              * DMS *',
     *                            2(I5,1H ,I2.2,1H',I2.2,1H",3H   *)/
     *'    * TIP RADIUS                         * MM  *', 2(F12.3, 3H   *)/
     *'    * STANDARD ADDENDUM                  * MM  *', 2(F12.3, 3H   *)/
     *'    * STANDARD DEDENDUM                  * MM  *', 2(F12.3, 3H   *)/
     *'    * PROTUBERANCE                       * MM  *', 2(F12.3, 3H   *))
10030 FORMAT ( '    ', 73(1H*) /
     *'    * DESIGN DETAILS                     *UNIT *     PINION    *',
     *'      WHEEL    *' / 3H    , 73(1H*) /
     *'    * NUMBER OF TEETH                    *     *', 2(I9,3X, 3H   *))
10031 FORMAT (
     *'    * HELIX ANGLE AT REFERENCE PCD       * DMS *',
     *                            2(I5,1H ,I2.2,1H',I2.2,1H",3H   *))
10032 FORMAT (
     *'    * ADDENDUM MODIFICATION COEFFICIENTS*     *', 2(F12.3, 3H   *)/
     *'    * RADIAL TOOTH TRUNCATIONS (TOPPING)* MM  *', 2(F12.3, 3H   *)/
     *'    * OPERATING NORMAL PRESSURE ANGLE    * DMS *',
     *                            2(I5,1H ,I2.2,1H',I2.2,1H",3H   *)/
     *'    * REFERENCE PITCH CIRCLE DIAMETERS   * MM  *', 2(F12.3, 3H   *)/
     *'    * OPERATING PITCH CIRCLE DIAMETERS   * MM  *', 2(F12.3, 3H   *)/
     *'    * OPERATING CENTRE DISTANCE          * MM  *', 2(F12.3, 3H   *)/
     *'    * OPERATING ADDENDA                  * MM  *', 2(F12.3, 3H   *)/
     *'    * OPERATING DEDENDA                  * MM  *', 2(F12.3, 3H   *)/
     *'    * ACTUAL DEPTH OF TEETH              * MM  *', 2(F12.3, 3H   *)/
     *'    * NOMINAL DEPTH OF TEETH             * MM  *', 2(F12.3, 3H   *)/
     *'    * INVOLUTE CLEARANCES                * MM  *', 2(F12.3, 3H   *)/
     *'    * BOTTOM CLEARANCES                  * MM  *', 2(F12.3, 3H   *)/
     *'    * SLIDE/ROLL RATIOS                  *     *', 2(F12.3, 3H   *)/
     *'    * CONTACT RATIO                      *     *', 2(F12.3, 3H   *))
10040 FORMAT (
     *'    * GEOMETRY FACTOR - PITTING          *     *', 2(F12.3, 3H   *)/
     *'    * GEOMETRY FACTORS - STRENGTH        *     *', 2(F12.3, 3H   *))
10050 FORMAT ( '    ', 73(1H*) /
     *'    * MACHINING DETAILS                  *UNIT *     PINION    *',
     *'      WHEEL    *' / 3H    , 73(1H*) /
     *'    * QUALITY NUMBERS (AGMA 390)         *     *', 2(I9,3X, 3H   *)/
     *'    * NORMAL THINNING OF TEETH(BACKLASH)* MM  *', 2(F12.3, 3H   *)/
     *'    * OUTSIDE DIAMETERS                  * MM  *', 2(F12.3, 3H   *)/
     *'    * ROOT DIA. INCLUDING BACKLASH       * MM  *', 2(F12.3, 3H   *)/
     *'    * RADII TO MID POINT OF INVOLUTE     * MM  *', 2(F12.3, 3H   *)/
     *'    * CHORDIAL HEIGHTS @ RMID            * MM  *', 2(F12.3, 3H   *)/
     *'    * CHORDIAL WIDTHS  @ RMID            * MM  *', 2(F12.3, 3H   *)/
     *'    * SPAN GAUGE DIMENSIONS              * MM  *', 2(F12.3, 3H   *)/
     *'    * NUMBER OF TEETH SPANNED            *     *', 2(I9,3X, 3H   *)/
     *'    * TOP LAND WIDTHS                    * MM  *', 2(F12.3, 3H   *)/
     *'    ', 73(1H*) /
```

```
     *'     * NOTE: CHORDAL & SPAN DIMENSIONS HAVE BACKLASH INCORPORATED',
     * 12X, '*' / '      ', 73(1H*) )
10060 FORMAT ( ///
     *'        DESIGNER: ____________________________          DATE : _________' )
C
  900 CONTINUE
        CLOSE ( 5 )
        CLOSE ( 6 )
        OPEN  ( 5, FILE='CON' )
        OPEN  ( 6, FILE='CON' )

      IF ( IOPT .EQ. 1 ) CALL PAUS

C     RESTORE ORIGINAL VALUES TO DLTARO, PHIC, PSIS
      DLTARO(1) = TDLTAR(1)
      DLTARO(2) = TDLTAR(2)
      PHIC = TPHIC
      PSIS = TPSIS

      RETURN
      END
```

APPENDIX B

TABULATION OF CONVERGENCE VALUES

BEFORE TESTS APPLIED								
MAXIMUM VALUES				MINIMUM VALUES				
NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM	K7
50	-0.1	0.000	2.000	10000000	0.3	0.500	-28.169	-1.571
1000	-0.3	0.400	119.387	500	-0.2	0.500	-58.341	-1.570
160	0.2	0.500	105.984	300	-0.4	0.500	-74.628	-1.560
200	-0.4	0.500	340.956	42	0.4	0.700	-269.251	-1.550
38	0.3	0.700	154.330	36	0.9	0.300	-366.750	-1.540
65	-0.1	0.600	69.649	85	-0.4	0.600	-1250.284	-1.530
75	-0.5	0.600	158.313	70	1.0	0.400	-271.889	-1.520
52	1.0	0.400	2160.056	14	0.3	0.900	-81.809	-1.510
40	1.0	0.400	707.646	44	-0.5	0.700	-53.613	-1.500
38	-0.5	0.700	116.666	30	1.0	0.400	-26.486	-1.490
26	1.0	0.400	99.574	24	-0.3	0.800	-60.183	-1.480
13	0.0	0.900	33.326	15	-0.1	0.900	-27.509	-1.470
18	1.0	0.300	194.012	15	-0.2	0.900	-31.759	-1.460
19	-0.4	0.800	32.233	15	1.0	0.300	-50.237	-1.450
19	-0.5	0.800	31.497	13	1.0	0.300	-56.059	-1.440
12	1.0	0.300	19.630	15	-0.5	0.900	-26.032	-1.430
16	-0.5	0.800	16.335	10	1.0	0.300	-51.056	-1.420
12	-0.4	0.900	17.651	10	-0.4	1.000	-20.893	-1.410
12	-0.5	0.900	23.137	10	-0.5	1.000	-19.269	-1.400
11	-0.5	0.900	17.328	10	-0.5	1.000	-5.501	-1.390
10	-0.5	0.900	9.771	10	-0.5	1.000	-1.212	-1.380
10	-0.5	0.900	10.848	10	-0.5	1.000	-0.358	-1.370
11	-0.5	0.800	7.065	10	-0.5	1.000	-0.067	-1.360
10	-0.5	0.800	6.468	500	-0.5	0.000	-0.003	-1.350
10	-0.5	0.800	5.753	500	-0.5	0.000	-0.003	-1.340
11	-0.5	0.700	4.872	500	-0.5	0.000	-0.003	-1.330
10	-0.5	0.700	4.760	500	-0.5	0.000	-0.003	-1.320
10	-0.5	0.700	4.278	500	-0.5	0.000	-0.003	-1.310
10	-0.5	0.600	4.050	500	-0.5	0.000	-0.003	-1.300
10	-0.5	0.600	3.845	400	-0.5	0.000	-0.004	-1.290
10	-0.5	0.500	3.579	400	-0.5	0.000	-0.004	-1.280
10	-0.5	0.500	3.456	400	-0.5	0.000	-0.004	-1.270
10	-0.5	0.400	3.247	400	-0.5	0.000	-0.004	-1.260
10	-0.5	0.400	3.144	400	-0.5	0.000	-0.004	-1.250
10	-0.5	0.300	2.993	400	-0.5	0.000	-0.004	-1.240
10	-0.5	0.300	2.888	400	-0.5	0.000	-0.005	-1.230
10	-0.5	0.200	2.776	400	-0.5	0.000	-0.005	-1.220
10	-0.5	0.100	2.673	400	-0.5	0.100	-0.005	-1.210
10	-0.5	0.000	2.571	400	-0.5	0.100	-0.005	-1.200
10	-0.5	0.000	2.455	300	-0.5	0.000	-0.005	-1.190

AFTER TESTS APPLIED AS PER ISO STANDARD TR 4467							
MAXIMUM VALUES				MINIMUM VALUES			
NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-
-	-	-	-	-	-	-	-

BEFORE TESTS APPLIED									AFTER TESTS APPLIED AS PER ISO STANDARD TR 4467							
MAXIMUM VALUES				MINIMUM VALUES					MAXIMUM VALUES				MINIMUM VALUES			
NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM	K7	NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM
10	-0.5	0.000	2.330	300	-0.5	0.000	-0.006	-1.180	-	-	-	-	-	-	-	-
10	-0.5	0.000	2.201	300	-0.5	0.000	-0.006	-1.170	-	-	-	-	-	-	-	-
10	-0.5	0.000	2.075	300	-0.5	0.000	-0.006	-1.160	-	-	-	-	-	-	-	-
10	-0.5	0.000	1.952	300	-0.5	0.000	-0.006	-1.150	-	-	-	-	-	-	-	-
10	-0.5	0.000	1.835	300	-0.5	0.000	-0.007	-1.140	-	-	-	-	-	-	-	-
10	-0.5	0.000	1.725	300	-0.5	0.000	-0.007	-1.130	-	-	-	-	-	-	-	-
10	-0.5	0.000	1.623	300	-0.5	0.000	-0.007	-1.120	-	-	-	-	-	-	-	-
10	-0.5	0.000	1.528	300	-0.5	0.000	-0.008	-1.110	-	-	-	-	-	-	-	-
10	-0.5	0.000	1.440	300	-0.5	0.000	-0.008	-1.100	-	-	-	-	-	-	-	-
10	-0.5	0.000	1.359	300	-0.5	0.000	-0.008	-1.090	-	-	-	-	-	-	-	-
10	-0.5	0.000	1.284	300	-0.5	0.100	-0.008	-1.080	-	-	-	-	-	-	-	-
10	-0.5	0.000	1.216	300	-0.5	0.100	-0.009	-1.070	-	-	-	-	-	-	-	-
10	-0.5	0.000	1.152	300	-0.5	0.100	-0.009	-1.060	-	-	-	-	-	-	-	-
10	-0.5	0.000	1.094	300	-0.5	0.100	-0.009	-1.050	-	-	-	-	-	-	-	-
10	-0.5	0.000	1.040	300	-0.5	0.100	-0.009	-1.040	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.991	300	-0.5	0.100	-0.010	-1.030	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.945	300	-0.5	0.100	-0.010	-1.020	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.903	300	-0.5	0.100	-0.010	-1.010	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.863	300	-0.5	0.100	-0.010	-1.000	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.827	200	-0.5	0.000	-0.011	-0.990	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.793	200	-0.5	0.000	-0.011	-0.980	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.762	200	-0.5	0.000	-0.012	-0.970	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.732	200	-0.5	0.000	-0.012	-0.960	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.705	200	-0.5	0.000	-0.013	-0.950	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.679	200	-0.5	0.000	-0.013	-0.940	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.655	200	-0.5	0.000	-0.014	-0.930	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.632	200	-0.5	0.000	-0.014	-0.920	-	-	-	-	-	-	-	-
42	-0.5	0.000	0.613	200	-0.5	0.000	-0.015	-0.910	-	-	-	-	-	-	-	-
42	-0.5	0.000	0.600	200	-0.5	0.000	-0.015	-0.900	-	-	-	-	-	-	-	-
42	-0.5	0.000	0.588	200	-0.5	0.000	-0.016	-0.890	-	-	-	-	-	-	-	-
42	-0.5	0.000	0.576	190	-0.5	0.000	-0.016	-0.880	-	-	-	-	-	-	-	-
42	-0.5	0.000	0.565	190	-0.5	0.000	-0.017	-0.870	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.554	190	-0.5	0.000	-0.017	-0.860	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.544	190	-0.5	0.000	-0.018	-0.850	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.534	180	-0.5	0.000	-0.018	-0.840	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.525	180	-0.5	0.000	-0.019	-0.830	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.516	180	-0.5	0.000	-0.019	-0.820	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.507	180	-0.5	0.000	-0.020	-0.810	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.499	170	-0.5	0.000	-0.020	-0.800	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.491	170	-0.5	0.000	-0.021	-0.790	-	-	-	-	-	-	-	-

BEFORE TESTS APPLIED									AFTER TESTS APPLIED AS PER ISO STANDARD TR 4467							
MAXIMUM VALUES				MINIMUM VALUES					MAXIMUM VALUES				MINIMUM VALUES			
NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM	K7	NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM
44	-0.5	0.000	0.483	170	-0.5	0.000	-0.022	-0.780	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.475	170	-0.5	0.000	-0.022	-0.770	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.467	170	-0.5	0.000	-0.023	-0.760	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.460	160	-0.5	0.000	-0.024	-0.750	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.453	160	-0.5	0.000	-0.024	-0.740	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.446	160	-0.5	0.000	-0.025	-0.730	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.440	160	-0.5	0.000	-0.026	-0.720	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.433	160	-0.5	0.000	-0.026	-0.710	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.427	150	-0.5	0.000	-0.027	-0.700	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.421	150	-0.5	0.000	-0.028	-0.690	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.414	150	-0.5	0.000	-0.029	-0.680	-	-	-	-	-	-	-	-
46	-0.5	0.000	0.409	150	-0.5	0.000	-0.029	-0.670	-	-	-	-	-	-	-	-
46	-0.5	0.000	0.403	150	-0.5	0.000	-0.030	-0.660	-	-	-	-	-	-	-	-
46	-0.5	0.000	0.397	140	-0.5	0.000	-0.031	-0.650	-	-	-	-	-	-	-	-
46	-0.5	0.000	0.392	140	-0.5	0.000	-0.032	-0.640	-	-	-	-	-	-	-	-
46	-0.5	0.000	0.386	140	-0.5	0.000	-0.033	-0.630	-	-	-	-	-	-	-	-
46	-0.5	0.000	0.381	140	-0.5	0.000	-0.033	-0.620	-	-	-	-	-	-	-	-
46	-0.5	0.000	0.376	140	-0.5	0.000	-0.034	-0.610	-	-	-	-	-	-	-	-
46	-0.5	0.000	0.370	140	-0.5	0.000	-0.035	-0.600	-	-	-	-	-	-	-	-
46	-0.5	0.000	0.365	140	-0.5	0.000	-0.036	-0.590	-	-	-	-	-	-	-	-
46	-0.5	0.000	0.360	130	-0.5	0.000	-0.037	-0.580	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.355	130	-0.5	0.000	-0.038	-0.570	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.350	130	-0.5	0.000	-0.039	-0.560	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.346	130	-0.5	0.000	-0.040	-0.550	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.341	130	-0.5	0.000	-0.041	-0.540	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.336	130	-0.5	0.000	-0.042	-0.530	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.332	130	-0.5	0.000	-0.043	-0.520	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.327	120	-0.5	0.000	-0.044	-0.510	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.323	120	-0.5	0.000	-0.046	-0.500	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.318	120	-0.5	0.000	-0.047	-0.490	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.314	120	-0.5	0.000	-0.048	-0.480	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.310	120	-0.5	0.000	-0.049	-0.470	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.305	120	-0.5	0.000	-0.050	-0.460	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.301	120	-0.5	0.000	-0.052	-0.450	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.297	120	-0.5	0.000	-0.053	-0.440	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.292	110	-0.5	0.000	-0.054	-0.430	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.288	110	-0.5	0.000	-0.056	-0.420	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.284	110	-0.5	0.000	-0.057	-0.410	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.280	110	-0.5	0.000	-0.059	-0.400	-	-	-	-	-	-	-	-
44	-0.5	0.000	0.276	110	-0.5	0.000	-0.060	-0.390	-	-	-	-	-	-	-	-

BEFORE TESTS APPLIED									AFTER TESTS APPLIED AS PER ISO STANDARD TR 4467							
MAXIMUM VALUES				MINIMUM VALUES					MAXIMUM VALUES				MINIMUM VALUES			
NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM	K7	NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM
44	-0.5	0.000	0.272	110	-0.5	0.000	-0.062	-0.380	-	-	-	-	-	-	-	-
42	-0.5	0.000	0.268	105	-0.5	0.000	-0.063	-0.370	-	-	-	-	-	-	-	-
42	-0.5	0.000	0.264	105	-0.5	0.000	-0.065	-0.360	-	-	-	-	-	-	-	-
42	-0.5	0.000	0.260	105	-0.5	0.000	-0.067	-0.350	-	-	-	-	-	-	-	-
42	-0.5	0.000	0.256	105	-0.5	0.000	-0.069	-0.340	-	-	-	-	-	-	-	-
42	-0.5	0.000	0.253	105	-0.5	0.000	-0.070	-0.330	-	-	-	-	-	-	-	-
42	-0.5	0.000	0.249	100	-0.5	0.000	-0.072	-0.320	-	-	-	-	-	-	-	-
42	-0.5	0.000	0.245	100	-0.5	0.000	-0.074	-0.310	-	-	-	-	-	-	-	-
42	-0.5	0.000	0.241	100	-0.5	0.000	-0.076	-0.300	-	-	-	-	-	-	-	-
42	-0.5	0.000	0.237	100	-0.5	0.000	-0.078	-0.290	-	-	-	-	-	-	-	-
42	-0.5	0.000	0.234	100	-0.5	0.000	-0.080	-0.280	-	-	-	-	-	-	-	-
40	-0.5	0.000	0.230	100	-0.5	0.000	-0.082	-0.270	-	-	-	-	-	-	-	-
40	-0.5	0.000	0.227	95	-0.5	0.000	-0.084	-0.260	-	-	-	-	-	-	-	-
40	-0.5	0.000	0.223	95	-0.5	0.000	-0.086	-0.250	-	-	-	-	-	-	-	-
40	-0.5	0.000	0.220	95	-0.5	0.000	-0.089	-0.240	-	-	-	-	-	-	-	-
40	-0.5	0.000	0.216	95	-0.5	0.000	-0.091	-0.230	-	-	-	-	-	-	-	-
40	-0.5	0.000	0.213	95	-0.5	0.000	-0.093	-0.220	13	0.6	0.300	0.136	13	0.6	0.702	0.077
40	-0.5	0.000	0.209	95	-0.5	0.000	-0.095	-0.210	13	0.3	0.000	0.155	13	0.6	0.702	0.077
38	-0.5	0.000	0.206	90	-0.5	0.000	-0.098	-0.200	13	0.3	0.000	0.152	13	0.6	0.702	0.076
38	-0.5	0.000	0.203	90	-0.5	0.000	-0.101	-0.190	14	0.3	0.100	0.151	13	0.6	0.702	0.076
38	-0.5	0.000	0.199	90	-0.5	0.000	-0.103	-0.180	15	0.2	0.000	0.158	13	0.6	0.702	0.075
38	-0.5	0.000	0.196	90	-0.5	0.000	-0.106	-0.170	16	0.2	0.000	0.156	13	0.6	0.702	0.075
38	-0.5	0.000	0.193	90	-0.5	0.000	-0.109	-0.160	17	0.2	0.100	0.154	13	0.6	0.702	0.074
38	-0.5	0.000	0.190	90	-0.5	0.000	-0.111	-0.150	18	0.1	0.000	0.159	19	0.6	0.000	0.064
36	-0.5	0.000	0.187	90	-0.5	0.000	-0.114	-0.140	19	0.1	0.000	0.156	20	0.6	0.000	0.051
36	-0.5	0.000	0.183	85	-0.5	0.000	-0.117	-0.130	20	0.0	0.000	0.161	22	0.7	0.100	0.011
36	-0.5	0.000	0.180	85	-0.5	0.000	-0.120	-0.120	22	0.0	0.000	0.158	24	0.6	0.000	0.005
36	-0.5	0.000	0.177	85	-0.5	0.000	-0.124	-0.110	22	0.0	0.100	0.155	24	0.6	0.000	0.000
34	-0.5	0.000	0.174	85	-0.5	0.000	-0.127	-0.100	26	-0.1	0.100	0.158	24	0.6	0.000	-0.006
34	-0.5	0.000	0.172	85	-0.5	0.000	-0.130	-0.090	26	-0.1	0.100	0.155	22	0.7	0.100	-0.010
34	-0.5	0.000	0.169	85	-0.5	0.000	-0.133	-0.080	32	-0.2	0.100	0.155	22	0.7	0.100	-0.015
34	-0.5	0.000	0.166	85	-0.5	0.000	-0.136	-0.070	40	-0.3	0.200	0.156	30	0.4	0.000	-0.008
32	-0.5	0.000	0.163	85	-0.5	0.000	-0.140	-0.060	44	-0.4	0.200	0.156	26	0.5	0.000	-0.010
32	-0.5	0.000	0.161	80	-0.5	0.000	-0.143	-0.050	50	-0.5	0.200	0.156	26	0.5	0.000	-0.016
32	-0.5	0.000	0.158	80	-0.5	0.000	-0.147	-0.040	50	-0.5	0.200	0.153	26	0.5	0.000	-0.022
32	-0.5	0.000	0.155	80	-0.5	0.000	-0.151	-0.030	50	-0.5	0.200	0.151	22	0.6	0.000	-0.025
30	-0.5	0.000	0.153	80	-0.5	0.000	-0.155	-0.020	50	-0.5	0.200	0.148	22	0.6	0.000	-0.031
30	-0.5	0.000	0.150	80	-0.5	0.000	-0.159	-0.010	50	-0.5	0.200	0.145	22	0.6	0.000	-0.036
10	-0.5	0.000	0.149	80	-0.5	0.000	-0.163	0.000	50	-0.5	0.200	0.142	28	0.4	0.000	-0.028
10	-0.5	0.000	0.147	80	-0.5	0.000	-0.167	0.010	50	-0.5	0.200	0.140	28	0.4	0.000	-0.033

BEFORE TESTS APPLIED								K7	AFTER TESTS APPLIED AS PER ISO STANDARD TR 4467							
MAXIMUM VALUES				MINIMUM VALUES					MAXIMUM VALUES				MINIMUM VALUES			
NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM	K7	NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM
10	-0.5	0.000	0.146	80	-0.5	0.000	-0.171	0.020	50	-0.5	0.200	0.137	26	0.7	0.200	-0.036
10	-0.5	0.000	0.144	80	-0.5	0.000	-0.174	0.030	50	-0.5	0.200	0.134	26	0.7	0.200	-0.041
10	-0.5	0.000	0.143	75	-0.5	0.000	-0.179	0.040	50	-0.5	0.200	0.132	24	0.5	0.000	-0.047
10	-0.5	0.000	0.141	75	-0.5	0.000	-0.183	0.050	50	-0.5	0.200	0.129	20	0.6	0.000	-0.051
10	-0.5	0.000	0.140	75	-0.5	0.000	-0.188	0.060	50	-0.5	0.200	0.126	20	0.6	0.000	-0.057
10	-0.5	0.000	0.138	75	-0.5	0.000	-0.193	0.070	50	-0.5	0.200	0.123	20	0.6	0.000	-0.062
10	-0.5	0.000	0.137	75	-0.5	0.000	-0.198	0.080	50	-0.5	0.200	0.120	19	0.6	0.000	-0.058
10	-0.5	0.000	0.135	75	-0.5	0.000	-0.203	0.090	50	-0.5	0.200	0.118	19	0.6	0.000	-0.064
10	-0.5	0.000	0.134	75	-0.5	0.000	-0.207	0.100	50	-0.5	0.200	0.115	19	0.6	0.000	-0.070
10	-0.5	0.000	0.133	75	-0.5	0.000	-0.212	0.110	50	-0.5	0.200	0.112	19	0.6	0.000	-0.076
10	-0.5	0.000	0.131	75	-0.5	0.000	-0.217	0.120	50	-0.5	0.200	0.109	19	0.6	0.000	-0.082
10	-0.5	0.000	0.130	75	-0.5	0.000	-0.222	0.130	90	-0.5	0.300	0.107	22	0.5	0.000	-0.079
10	-0.5	0.000	0.128	75	-0.5	0.000	-0.226	0.140	85	-0.5	0.300	0.105	20	0.6	0.100	-0.084
10	-0.5	0.000	0.127	75	-0.5	0.000	-0.231	0.150	80	-0.5	0.300	0.103	18	0.6	0.000	-0.089
10	-0.5	0.000	0.125	70	-0.5	0.000	-0.237	0.160	80	-0.5	0.300	0.102	18	0.6	0.000	-0.095
10	-0.5	0.000	0.124	70	-0.5	0.000	-0.243	0.170	75	-0.5	0.300	0.100	19	0.6	0.100	-0.092
10	-0.5	0.000	0.122	70	-0.5	0.000	-0.249	0.180	50	-0.5	0.300	0.099	19	0.6	0.100	-0.098
10	-0.5	0.000	0.120	70	-0.5	0.000	-0.255	0.190	50	-0.5	0.300	0.097	17	0.6	0.000	-0.102
10	-0.5	0.000	0.119	70	-0.5	0.000	-0.261	0.200	50	-0.5	0.300	0.096	17	0.6	0.000	-0.108
10	-0.5	0.000	0.117	70	-0.5	0.000	-0.267	0.210	50	-0.5	0.300	0.095	17	0.6	0.000	-0.114
10	-0.5	0.000	0.116	70	-0.5	0.000	-0.273	0.220	50	-0.5	0.300	0.093	20	0.5	0.000	-0.112
10	-0.5	0.000	0.114	70	-0.5	0.000	-0.279	0.230	50	-0.5	0.300	0.092	16	0.6	0.000	-0.115
10	-0.5	0.000	0.113	70	-0.5	0.000	-0.285	0.240	50	-0.5	0.300	0.091	16	0.6	0.000	-0.121
10	-0.5	0.000	0.111	70	-0.5	0.000	-0.291	0.250	50	-0.5	0.300	0.089	16	0.6	0.000	-0.127
10	-0.5	0.000	0.109	70	-0.5	0.000	-0.297	0.260	50	-0.5	0.300	0.088	17	0.6	0.100	-0.125
10	-0.5	0.000	0.108	70	-0.5	0.000	-0.303	0.270	50	-0.5	0.300	0.086	15	0.6	0.000	-0.128
10	-0.5	0.000	0.106	70	-0.5	0.000	-0.308	0.280	50	-0.5	0.300	0.085	15	0.6	0.000	-0.134
10	-0.5	0.000	0.104	70	-0.5	0.000	-0.314	0.290	50	-0.5	0.300	0.084	15	0.6	0.000	-0.140
10	-0.5	0.000	0.102	70	-0.5	0.000	-0.320	0.300	50	-0.5	0.300	0.082	16	0.6	0.100	-0.139
10	-0.5	0.000	0.101	65	-0.5	0.000	-0.327	0.310	50	-0.5	0.300	0.081	14	0.6	0.000	-0.140
10	-0.5	0.000	0.099	65	-0.5	0.000	-0.335	0.320	50	-0.5	0.300	0.079	14	0.6	0.000	-0.147
10	-0.5	0.000	0.097	65	-0.5	0.000	-0.343	0.330	50	-0.5	0.300	0.078	14	0.6	0.000	-0.153
10	-0.5	0.000	0.095	65	-0.5	0.000	-0.350	0.340	50	-0.5	0.300	0.076	17	0.5	0.000	-0.150
10	-0.5	0.000	0.093	65	-0.5	0.000	-0.358	0.350	50	-0.5	0.300	0.075	50	-0.5	0.000	-0.155
10	-0.5	0.000	0.091	65	-0.5	0.000	-0.366	0.360	50	-0.5	0.300	0.073	13	0.6	0.000	-0.159
10	-0.5	0.100	0.089	65	-0.5	0.000	-0.373	0.370	50	-0.5	0.300	0.072	14	0.6	0.100	-0.160
10	-0.5	0.100	0.087	65	-0.5	0.000	-0.381	0.380	50	-0.5	0.300	0.070	46	-0.4	0.000	-0.166
10	-0.5	0.200	0.085	65	-0.5	0.000	-0.389	0.390	50	-0.5	0.300	0.069	46	-0.3	0.100	-0.163
10	-0.5	0.200	0.083	65	-0.5	0.000	-0.396	0.400	50	-0.5	0.300	0.067	42	-0.3	0.000	-0.169
10	-0.5	0.200	0.081	65	-0.5	0.000	-0.404	0.410	50	-0.5	0.300	0.066	42	-0.3	0.000	-0.177

BEFORE TESTS APPLIED									AFTER TESTS APPLIED AS PER ISO STANDARD TR 4467							
MAXIMUM VALUES				MINIMUM VALUES					MAXIMUM VALUES				MINIMUM VALUES			
NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM	K7	NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM
10	-0.5	0.300	0.080	65	-0.5	0.000	-0.412	0.420	50	-0.5	0.300	0.064	15	0.5	0.000	-0.176
10	-0.5	0.300	0.078	65	-0.5	0.000	-0.420	0.430	50	-0.5	0.300	0.063	38	-0.2	0.000	-0.180
10	-0.5	0.300	0.076	65	-0.5	0.000	-0.427	0.440	50	-0.5	0.300	0.061	38	-0.2	0.000	-0.188
10	-0.5	0.300	0.074	65	-0.5	0.000	-0.435	0.450	50	-0.5	0.383	0.060	44	-0.4	0.000	-0.189
10	-0.5	0.300	0.072	65	-0.5	0.000	-0.443	0.460	50	-0.5	0.383	0.059	44	-0.4	0.000	-0.198
10	-0.5	0.400	0.070	65	-0.5	0.000	-0.450	0.470	50	-0.5	0.383	0.058	44	-0.3	0.100	-0.196
11	-0.5	0.400	0.068	65	-0.5	0.000	-0.458	0.480	50	-0.5	0.383	0.058	40	-0.3	0.000	-0.201
11	-0.5	0.400	0.066	65	-0.5	0.000	-0.466	0.490	50	-0.5	0.383	0.057	40	-0.3	0.000	-0.210
12	-0.5	0.400	0.065	60	-0.5	0.000	-0.476	0.500	50	-0.5	0.383	0.056	50	-0.5	0.100	-0.206
13	-0.5	0.400	0.063	60	-0.5	0.000	-0.486	0.510	50	-0.5	0.383	0.055	50	-0.5	0.100	-0.216
13	-0.5	0.400	0.061	60	-0.5	0.000	-0.497	0.520	50	-0.5	0.383	0.054	46	-0.4	0.100	-0.214
14	-0.5	0.400	0.060	60	-0.5	0.000	-0.507	0.530	50	-0.5	0.383	0.053	46	-0.4	0.100	-0.224
15	-0.5	0.400	0.058	60	-0.5	0.000	-0.518	0.540	50	-0.5	0.383	0.053	32	-0.1	0.000	-0.223
14	-0.5	0.500	0.057	60	-0.5	0.000	-0.528	0.550	50	-0.5	0.383	0.052	38	-0.3	0.000	-0.225
14	-0.5	0.500	0.056	60	-0.5	0.000	-0.539	0.560	50	-0.5	0.383	0.051	38	-0.3	0.000	-0.235
15	-0.5	0.500	0.054	60	-0.5	0.000	-0.550	0.570	50	-0.5	0.383	0.050	38	-0.3	0.000	-0.245
15	-0.5	0.500	0.053	60	-0.5	0.000	-0.561	0.580	50	-0.5	0.383	0.049	34	-0.2	0.000	-0.237
16	-0.5	0.500	0.052	60	-0.5	0.000	-0.571	0.590	50	-0.5	0.383	0.048	34	-0.2	0.000	-0.247
17	-0.5	0.500	0.051	60	-0.5	0.000	-0.582	0.600	50	-0.5	0.383	0.047	40	-0.3	0.100	-0.242
18	-0.5	0.500	0.050	60	-0.5	0.000	-0.593	0.610	50	-0.5	0.383	0.046	36	-0.2	0.100	-0.241
18	-0.5	0.500	0.048	60	-0.5	0.000	-0.605	0.620	50	-0.5	0.383	0.045	26	0.0	0.000	-0.242
19	-0.5	0.500	0.047	60	-0.5	0.000	-0.616	0.630	50	-0.5	0.383	0.044	32	-0.2	0.000	-0.246
20	-0.5	0.500	0.046	58	-0.5	0.000	-0.628	0.640	50	-0.5	0.383	0.043	32	-0.2	0.000	-0.256
20	-0.5	0.500	0.045	58	-0.5	0.000	-0.641	0.650	50	-0.5	0.383	0.042	32	-0.2	0.000	-0.267
22	-0.5	0.500	0.044	58	-0.5	0.000	-0.654	0.660	50	-0.5	0.383	0.041	38	-0.3	0.100	-0.265
22	-0.5	0.500	0.043	58	-0.5	0.000	-0.667	0.670	50	-0.5	0.383	0.040	34	-0.2	0.100	-0.264
24	-0.5	0.500	0.042	58	-0.5	0.000	-0.681	0.680	50	-0.5	0.383	0.039	26	0.0	0.100	-0.252
24	-0.5	0.500	0.041	58	-0.5	0.000	-0.694	0.690	50	-0.5	0.383	0.038	60	-0.5	0.200	-0.255
24	-0.5	0.500	0.040	58	-0.5	0.000	-0.708	0.700	50	-0.5	0.383	0.037	26	-0.1	0.000	-0.263
26	-0.5	0.500	0.039	58	-0.5	0.000	-0.721	0.710	50	-0.5	0.383	0.036	26	-0.1	0.000	-0.274
26	-0.5	0.500	0.038	56	-0.5	0.000	-0.737	0.720	50	-0.5	0.383	0.035	32	-0.2	0.100	-0.281
28	-0.5	0.500	0.037	56	-0.5	0.000	-0.753	0.730	50	-0.5	0.383	0.034	58	-0.5	0.200	-0.268
30	-0.5	0.500	0.036	56	-0.5	0.000	-0.769	0.740	50	-0.5	0.383	0.032	20	0.0	0.000	-0.258
30	-0.5	0.500	0.035	56	-0.5	0.000	-0.786	0.750	50	-0.5	0.383	0.031	20	0.0	0.000	-0.270
32	-0.5	0.500	0.034	56	-0.5	0.000	-0.803	0.760	50	-0.5	0.383	0.030	26	-0.1	0.100	-0.280
32	-0.5	0.500	0.034	56	-0.5	0.000	-0.820	0.770	50	-0.5	0.383	0.029	50	-0.4	0.200	-0.263
34	-0.5	0.500	0.033	56	-0.5	0.000	-0.837	0.780	50	-0.5	0.383	0.027	50	-0.4	0.200	-0.274
34	-0.5	0.500	0.032	54	-0.5	0.000	-0.855	0.790	54	-0.5	0.381	0.026	54	-0.5	0.200	-0.279
36	-0.5	0.500	0.031	54	-0.5	0.000	-0.876	0.800	58	-0.5	0.379	0.025	54	-0.5	0.200	-0.291
38	-0.5	0.500	0.030	54	-0.5	0.000	-0.897	0.810	65	-0.5	0.376	0.024	48	-0.4	0.200	-0.282

BEFORE TESTS APPLIED								K7	AFTER TESTS APPLIED AS PER ISO STANDARD TR 4467							
MAXIMUM VALUES				MINIMUM VALUES					MAXIMUM VALUES				MINIMUM VALUES			
NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM	K7	NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM
38	-0.5	0.500	0.030	54	-0.5	0.000	-0.919	0.820	70	-0.5	0.374	0.022	52	-0.5	0.200	-0.288
40	-0.5	0.500	0.029	54	-0.5	0.000	-0.941	0.830	75	-0.5	0.373	0.021	52	-0.5	0.200	-0.301
42	-0.5	0.500	0.028	54	-0.5	0.000	-0.963	0.840	85	-0.5	0.370	0.020	46	-0.4	0.200	-0.293
44	-0.5	0.500	0.027	54	-0.5	0.000	-0.986	0.850	95	-0.5	0.368	0.018	50	-0.5	0.200	-0.300
44	-0.5	0.500	0.027	52	-0.5	0.000	-1.011	0.860	110	-0.5	0.365	0.017	44	-0.4	0.200	-0.293
46	-0.5	0.500	0.026	52	-0.5	0.000	-1.038	0.870	130	-0.5	0.363	0.016	38	-0.3	0.200	-0.289
48	-0.5	0.500	0.025	52	-0.5	0.000	-1.067	0.880	150	-0.5	0.361	0.015	110	-0.4	0.300	-0.203
50	-0.5	0.500	0.025	52	-0.5	0.000	-1.096	0.890	180	-0.5	0.359	0.014	120	-0.5	0.300	-0.204
52	-0.5	0.500	0.024	52	-0.5	0.000	-1.126	0.900	200	-0.5	0.358	0.012	105	-0.4	0.300	-0.199
54	-0.5	0.500	0.023	52	-0.5	0.000	-1.157	0.910	300	-0.5	0.355	0.011	90	-0.3	0.300	-0.196
54	-0.5	0.500	0.023	50	-0.5	0.000	-1.190	0.920	400	-0.5	0.354	0.010	100	-0.4	0.300	-0.197
56	-0.5	0.500	0.022	50	-0.5	0.000	-1.228	0.930	1000	-0.5	0.351	0.009	100	-0.4	0.300	-0.205
58	-0.5	0.500	0.021	50	-0.5	0.000	-1.267	0.940	5000	-0.5	0.349	0.008	110	-0.5	0.300	-0.207
60	-0.5	0.500	0.021	50	-0.5	0.000	-1.307	0.950	10000000	0.9	0.349	0.007	95	-0.4	0.300	-0.205
65	-0.5	0.500	0.020	50	-0.5	0.000	-1.349	0.960	10000000	0.9	0.349	0.006	105	-0.5	0.300	-0.207
65	-0.5	0.500	0.020	50	-0.5	0.000	-1.392	0.970	10000000	0.9	0.349	0.005	90	-0.4	0.300	-0.206
65	-0.5	0.500	0.019	48	-0.5	0.000	-1.443	0.980	10000000	0.9	0.349	0.004	100	-0.5	0.300	-0.209
70	-0.5	0.500	0.018	48	-0.5	0.000	-1.496	0.990	10000000	0.9	0.349	0.003	70	-0.3	0.300	-0.203
70	-0.5	0.500	0.018	10	-0.5	0.000	-1.559	1.000	10000000	0.9	0.349	0.002	95	-0.5	0.300	-0.213
75	-0.5	0.500	0.017	10	-0.5	0.000	-1.805	1.010	10000000	0.9	0.349	0.001	90	-0.5	0.300	-0.208
52	-0.5	0.600	0.017	10	-0.5	0.000	-2.103	1.020	10000000	0.9	0.349	0.000	58	-0.3	0.300	-0.208
80	-0.5	0.500	0.016	10	-0.5	0.000	-2.465	1.030	10000000	0.9	0.349	-0.001	85	-0.5	0.300	-0.217
54	-0.5	0.600	0.016	10	-0.5	0.000	-2.905	1.040	10000000	0.9	0.349	-0.002	80	-0.5	0.300	-0.216
56	-0.5	0.600	0.015	10	-0.5	0.000	-3.437	1.050	10000000	0.9	0.349	-0.003	75	-0.5	0.300	-0.217
58	-0.5	0.600	0.015	10	-0.5	0.000	-4.069	1.060	10000000	0.9	0.349	-0.005	44	-0.4	0.300	-0.225
60	-0.5	0.600	0.014	10	-0.5	0.000	-4.795	1.070	10000000	0.9	0.349	-0.006	65	-0.5	0.300	-0.227
65	-0.5	0.600	0.014	10	-0.5	0.000	-5.575	1.080	10000000	0.9	0.349	-0.007	50	-0.5	0.300	-0.232
65	-0.5	0.600	0.013	10	-0.5	0.000	-6.312	1.090	10000000	0.9	0.349	-0.008	32	-0.1	0.400	-0.117
70	-0.5	0.600	0.013	10	-0.5	0.000	-6.850	1.100	10000000	0.9	0.349	-0.009	32	-0.2	0.400	-0.123
70	-0.5	0.600	0.013	10	-0.5	0.000	-7.017	1.110	10000000	0.9	0.349	-0.010	34	-0.2	0.400	-0.121
75	-0.5	0.600	0.012	11	-0.5	0.000	-6.939	1.120	10000000	0.9	0.349	-0.011	38	-0.1	0.400	-0.112
75	-0.5	0.600	0.012	11	-0.5	0.000	-7.049	1.130	10000000	0.9	0.349	-0.012	38	-0.3	0.400	-0.117
80	-0.5	0.600	0.011	12	-0.5	0.000	-7.077	1.140	10000000	0.9	0.349	-0.013	40	-0.3	0.400	-0.117
80	-0.5	0.600	0.011	12	-0.5	0.100	-7.076	1.150	10000000	0.9	0.349	-0.015	44	-0.2	0.400	-0.110
85	-0.5	0.600	0.010	13	-0.5	0.100	-7.122	1.160	10000000	0.9	0.349	-0.016	50	-0.5	0.383	-0.119
130	-0.5	0.500	0.010	14	-0.5	0.100	-7.133	1.170	10000000	0.9	0.349	-0.017	50	-0.5	0.383	-0.132
140	-0.5	0.500	0.010	15	-0.5	0.100	-7.151	1.180	10000000	0.9	0.349	-0.018	56	-0.5	0.380	-0.128
140	-0.5	0.500	0.009	14	-0.5	0.200	-7.187	1.190	10000000	0.9	0.349	-0.019	65	-0.5	0.376	-0.120
150	-0.5	0.500	0.009	15	-0.5	0.200	-7.264	1.200	10000000	0.9	0.349	-0.020	70	-0.5	0.374	-0.122
160	-0.5	0.500	0.009	16	-0.5	0.200	-7.329	1.210	10000000	0.9	0.349	-0.021	80	-0.5	0.371	-0.118

BEFORE TESTS APPLIED									AFTER TESTS APPLIED AS PER ISO STANDARD TR 4467							
MAXIMUM VALUES				MINIMUM VALUES					MAXIMUM VALUES				MINIMUM VALUES			
NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM	K7	NE	X	PHIL	MAXIMUM	NE	X	PHIL	MINIMUM
160	-0.5	0.500	0.008	17	-0.5	0.200	-7.378	1.220	10000000	0.9	0.349	-0.023	90	-0.5	0.369	-0.116
170	-0.5	0.500	0.008	19	-0.5	0.200	-7.441	1.230	10000000	0.9	0.349	-0.024	100	-0.5	0.367	-0.116
180	-0.5	0.500	0.008	20	-0.5	0.200	-7.497	1.240	10000000	0.9	0.349	-0.025	120	-0.5	0.364	-0.109
190	-0.5	0.500	0.007	18	-0.5	0.300	-7.593	1.250	10000000	0.9	0.349	-0.026	150	-0.5	0.361	-0.099
200	-0.5	0.500	0.007	19	-0.5	0.300	-7.756	1.260	10000000	0.9	0.349	-0.027	180	-0.5	0.359	-0.095
200	-0.5	0.500	0.007	20	-0.5	0.300	-7.888	1.270	10000000	0.9	0.349	-0.028	300	-0.5	0.355	-0.072
10	-0.5	0.000	0.055	22	-0.5	0.300	-8.079	1.280	10000000	0.9	0.349	-0.030	400	-0.5	0.354	-0.066
10	-0.5	0.000	0.132	24	-0.5	0.300	-8.243	1.290	10000000	0.9	0.349	-0.031	500	-0.1	0.357	-0.048
10	-0.5	0.000	0.202	26	-0.5	0.300	-8.433	1.300	10000000	0.9	0.349	-0.032	2000	-0.5	0.350	-0.041
10	-0.5	0.000	0.267	28	-0.5	0.300	-8.638	1.310	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.330	30	-0.5	0.300	-8.812	1.320	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.391	34	-0.5	0.300	-9.053	1.330	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.453	38	-0.5	0.300	-9.264	1.340	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.517	42	-0.5	0.300	-9.477	1.350	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.587	48	-0.5	0.300	-9.671	1.360	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.663	34	-0.5	0.400	-9.992	1.370	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.750	36	-0.5	0.400	-10.560	1.380	-	-	-	-	-	-	-	-
10	-0.5	0.000	0.848	10	1.0	0.300	-12.766	1.390	-	-	-	-	-	-	-	-
10	1.0	0.300	100.311	11	1.0	0.300	-17.442	1.400	-	-	-	-	-	-	-	-
11	1.0	0.300	32.651	12	1.0	0.300	-36.651	1.410	-	-	-	-	-	-	-	-
13	1.0	0.300	102.383	14	1.0	0.300	-26.643	1.420	-	-	-	-	-	-	-	-
15	1.0	0.300	88.580	16	1.0	0.300	-35.857	1.430	-	-	-	-	-	-	-	-
17	1.0	0.400	34.657	18	1.0	0.400	-336.049	1.440	-	-	-	-	-	-	-	-
20	1.0	0.400	31.646	22	1.0	0.400	-55.424	1.450	-	-	-	-	-	-	-	-
24	1.0	0.400	30.984	26	1.0	0.400	-161.833	1.460	-	-	-	-	-	-	-	-
30	1.0	0.400	41.807	32	1.0	0.400	-179.850	1.470	-	-	-	-	-	-	-	-
38	1.0	0.400	46.118	40	1.0	0.400	-1026.894	1.480	-	-	-	-	-	-	-	-
50	1.0	0.400	56.496	52	1.0	0.400	-2835.906	1.490	-	-	-	-	-	-	-	-
70	1.0	0.400	290.439	170	-0.5	0.400	-27.596	1.500	-	-	-	-	-	-	-	-
100	1.0	0.400	150.018	24	0.9	0.300	-31.748	1.510	-	-	-	-	-	-	-	-
36	0.9	0.300	527.363	160	1.0	0.300	-33.302	1.520	-	-	-	-	-	-	-	-
60	0.9	0.300	18.540	400	-0.5	0.400	-39.650	1.530	-	-	-	-	-	-	-	-
17	1.0	0.700	23.482	140	0.9	0.300	-55.444	1.540	-	-	-	-	-	-	-	-
44	1.0	0.600	95.181	1000	-0.3	0.400	-68.027	1.550	-	-	-	-	-	-	-	-
400	0.9	0.400	25.045	200	0.9	0.500	-67.375	1.560	-	-	-	-	-	-	-	-
90	0.7	0.500	61.823	1400	-0.5	0.400	-339.306	1.570	-	-	-	-	-	-	-	-
44	-0.1	0.000	2.000	10000000	0.3	0.500	-28.169	1.571	-	-	-	-	-	-	-	-

APPENDIX C

README FILE FOR GEARGEOM FLOPPY DISC

GEAR GEOMETRY PROGRAM

"GEARGEOM.EXE"

GEAR GEOMETRY FACTOR PROGRAM

ASSP-001-1987

VERSION 1.00

JUNE 1987

DESIGNED BY :- R. DAVEY

PROGRAMMED BY :- E. HUTTON

GEAR GEOMETRY PROGRAM

TABLE OF CONTENTS.

GEAR GEOMETRY PROGRAM.

GEAR GEOMETRY FACTORS PROGRAM ASSP-001-1987 "GEARGEOM.EXE"

PROGRAM SPECIFICATION.

1. This program calculates the geometry factors for pitting resistance (I) and bending strength (J) for parallel shaft external spur and helical involute gears, in accordance with AGMA 218.01 and AS 2938-1987.

2. The calculations are based on gear teeth manufactured by a hob or a rack cutter without protuberance and do not include manufacturing tolerances, which have only a marginal effect on the calculated values of (I) and (J).

3. The program is universal and as such has inherent dangers for the inexperienced gear designer. The program automatically identifies and suggests solutions to four of these potential errors. Namely that of tip to root fillet interference, insufficient bottom clearance, insufficient top land width and non conjugate action.

4. The program was designed by R. Davey and programmed by E. Hutton of BHP Steel International Group Slab and Plate Products Division Port Kembla Steelworks and donated to SAA by that company.

5. The program is written in Fortran 77 and compiled using Microsoft Fortran Compiler Version 3.31 and will run on IBM PC/XT/AT or compatible with a minimum of 256 KBytes of RAM memory, and one double sided floppy diskette drive.

6. SAA obtained copies of the diskette from FIGTREE COMPUTER SERVICES PTY. LTD. of 116 Montague Street, North Wollongong (Phone 042-284333), which is also responsible for maintaining the program. All enquiries concerning the operation of the program should be directed to that company, except those relating to copyright which should be directed to SAA. (Phone 02-9634223)

7. A supplementary program to GEARGEOM, IRATE, pertaining to the allowable transmitted power rating as per AGMA 218.01 is available from E. Hutton or R Davey.(Phone 042-291282)

GEAR GEOMETRY PROGRAM.

LIMITATION OF LIABILITIES AND DISCLAIMER OF WARRANTIES.

1. Only limited validation of the program has been done by SAA and users are advised to satisfy themselves that the program is suitable for their purposes. As SAA was not directly involved in the design or manufacture of the program, it is necessary to state that it is supplied on the strict understanding that neither SAA nor any of its officers are to incur any legal liability whatsoever (including liability for negligence) should the software incorrectly apply the material in the standard or be incomplete or in some other way defective and all liability must therefore be disclaimed.

2. BHP Steel International Group Slab and Plate Products Division Port Kembla Steelworks and Figtree Computer Services Pty. Ltd. make no expressed or implied warranties of any kind with regard to performance or fitness of purpose for the program. In no event shall BHP Steel International Group Slab and Plate Products Division Port Kembla Steelworks and/or Figtree Computer Services Pty. Ltd. or any of their employees be liable for incidental or consequential damages in connection with or arising out of the distribution or use of the program.

GEAR GEOMETRY PROGRAM.

1. GEAR GEOMETRY FACTORS "I" AND "J".

The calculation of gear geometry factors as defined in AGMA 218.01 and Appendix D of Australian Standard AS 2938-1987 necessitates numerous computations. This program reduces the tedium, thus enabling the analysis of various solutions to the same problem. The results can be displayed on the screen, printed by a standard printer or stored to disk for later processing.

2. WHAT YOU NEED TO RUN GEARGEOM.

To successfully calculate the geometry factors for a gear set on your IBM PC/XT/AT or compatible, you need:

Your GEARGEOM diskette.

At least 256K bytes of random access memory (RAM).

One diskette drive

A Display monitor.

A Printer (optional but recommended).

Either PC or MS DOS diskette, Version 2.0 or later.

Blank 5-1/4 inch diskettes

3. DISKETTE PROTECTION.

Before using the GEARGEOM program you should, for your own protection, either make a work-copy of the distribution diskette or copy GEARGEOM onto your hard disk then store the original safely away.

To make a work-copy of the diskette you must first format a blank diskette. Use the DOS FORMAT command with the S option. This formats the diskette and copies the DOS system files and COMMAND.COM onto your diskette. You can start DOS using this diskette. Next use the DOS COPY command to copy the master files to your work-copy diskette. Refer to the IBM Personal Computer Disk Operating System manual for more information about formatting and copying.

To copy GEARGEOM onto your hard disk you may either create a subdirectory on your hard disk or use an existing directory, then use the DOS COPY command to copy all the files on the original diskette to your hard disk. Refer to the IBM Personal Computer Disk Operating System manual for more information about subdirectories and copying.

GEAR GEOMETRY PROGRAM.

It should be noted that any enquiries concerning the operation of GEARGEOM, will only be answered by FIGTREE COMPUTER SERVICES PTY. LTD., if the enquirer is registered as having purchased the original software from SAA.

4. FILES ON THE DISKETTE.

The diskette contains the following files.

GEARGEOM.EXE	Gear geometry factor calculation program
GEOMLOGO.TXT	Compulsory file.
README	GEARGEOM program documentation file. (you are reading it)
	(Use the following DOS command to obtain a hard copy of this file on your printer.

COPY README PRN

The following example data sets are for the worked examples in Appendix D of AS2938-1987.

EXAMPLE1.DAT	EXAMPLE NO. B3.2 WITH BACKLASH & TRUNCATION
EXAMPLE2.DAT	EXAMPLE NO. D3.1.2 WITH BACKLASH (FIG. B2 AGMA 218.01)
EXAMPLE3.DAT	EXAMPLE NO. D3.1.3 WITH BACKLASH (FIG. B2 AGMA 218.01). (INACCURATE)
EXAMPLE4.DAT	EXAMPLE NO. D3.1.4 WITH BACKLASH (FIG. B1 AGMA 218.01).
EXAMPLE5.DAT	EXAMPLE NO. D3.1.5 WITH BACKLASH (FIG. B1 AGMA 218.01)
EXAMPLE6.DAT	EXAMPLE NO. B4.2 WITH BACKLASH
EXAMPLE7.DAT	EXAMPLE NO. D3.2.2 WITH BACKLASH NO TRUNCATION
EXAMPLE8.DAT	EXAMPLE N0. D3.2.2 WITH BACKLASH AND TRUNCATION
EXAMPLE9.DAT	EXAMPLE NO. B4.2 WITH BACKLASH.

GEAR GEOMETRY PROGRAM.

5. METHOD OF OPERATION.

To use the program, place the diskette in the drive. Make sure that the drive is the default drive, then commence execution by typing GEARGEOM at the DOS prompt.

A:>GEARGEOM

After the preamble has been displayed, the INITIAL DATA FILE SELECTION menu is displayed

```
******************************************************************
* GEAR GEOMETRY PROGRAM ASSP-001-1987          Ver 1.00 JUN 1987 *
******************************************************************
*                                                                *
* CALCULATION OF THE GEOMETRY FACTORS FOR PITTING RESISTANCE AND *
* BENDING STRENGTH FOR EXTERNAL SPUR AND HELICAL INVOLUTE GEAR   *
* TEETH WITHOUT PROTUBERANCE                                     *
*                                                                *
*                                                                *
* GEAR GEOMETRY PROGRAM   MODE :- INITIAL SET OF DATA            *
*                                                                *
*                                                                *
* PLEASE SELECT THE REQUIRED OPTION                              *
*                                                                *
* 1 = CREATE                                                     *
* 2 = RETRIEVE                                                   *
*                                                                *
* 9 = EXIT FROM PROGRAM                                          *
*                                                                *
******************************************************************

SELECT OPTION  ==>
```

Selection of "1" allows for the creation of a new set of data, in order to calculate the geometry factors. (Refer Section 6.1)

Selection of "2" enables the set of data to be initialised from a stored data file, in order to calculate the geometry factors. (Refer Section 6.2)

Selection of "9" will terminate the program.

After either option "1" or "2" has been executed, the SELECTED SET OF DATA menu is displayed.

GEAR GEOMETRY PROGRAM.

```
*******************************************************************************
* GEAR GEOMETRY PROGRAM ASSP-001-1987                   Ver 1.00 JUN 1987    *
*******************************************************************************
*                                                                             *
* CALCULATION OF THE GEOMETRY FACTORS FOR PITTING RESISTANCE AND             *
* BENDING STRENGTH FOR EXTERNAL SPUR AND HELICAL INVOLUTE GEAR               *
* TEETH WITHOUT PROTUBERANCE                                                  *
*                                                                             *
*                                                                             *
* GEAR GEOMETRY PROGRAM   MODE :- SELECTED SET OF DATA                        *
* FILE NAME = ...none...                                                      *
*                                                                             *
*                                                                             *
*                                                                             *
* PLEASE SELECT THE REQUIRED OPTION FOR THE CURRENT SET OF DATA              *
*                                                                             *
* 1 = ANALYSE                                                                 *
* 2 = EDIT                                                                    *
* 3 = SAVE                                                                    *
*                                                                             *
* 9 = RETURN to INITIAL SET OF DATA MENU                                      *
*                                                                             *
*******************************************************************************

SELECT OPTION  ==>
```

Selection of "1" analyses the current gear set data, calculating the I and J geometry factors. The calculations can be displayed on the screen, printed by a printer or stored to a file. (Refer Section 6.4)

Selection of "2" allows for editing the current gear set data. (Refer Section 6.5)

Selection of "3" allows the current gear set data file to be stored to disk. (Refer Section 6.3)

Selection "9" will return to the previous menu.

6. SAMPLE SESSION

This section uses a demonstration gear set to illustrate the step by step instructions for using the GEARGEOM program.

If you enter commands exactly as described in this section, you will have a successful session with the GEARGEOM program.

The six steps in analysing a gear set are:

1. Create a set of data or retrieve a stored data file.

2. View and verify the set of data.

GEAR GEOMETRY PROGRAM.

3. Save the set of data to a data file (optional).

4. Analyse the set of data.

5. Edit - Change any values in the set of data that may need correction as a result of the analysis.

6. Print out the results of the analysis.

Data input in the create or edit mode.

The basic symbols are divided into letters, digits, and special symbols.

LETTERS	A to Z, a to z
DIGITS	0 1 2 3 4 5 6 7 8 9
SPECIAL SYMBOLS	+ - * / = ^ ( ) [ ] { } , : ; ' $! @ % &

Data types used in GEARGEOM are CHARACTERS, INTEGERS and REALS.

CHARACTERS are composed of LETTERS, DIGITS or SPECIAL SYMBOLS

INTEGERS are whole numbers in the range of -2 147 483 637 to 2 147 483 647 and are composed of DIGITS with an optional leading negative sign.

REALS are fractional numbers that range from 4.19D-307 1.67D+308, -1.67D+308 to -4.19D-307, and zero with up to 15 significant digits, and are composed with DIGITS with with an optional decimal point, an optional leading negative sign and an optional exponent.

NOTE :- Where two variables are requested, then the two values must be separated either with a comma or a carriage return. i.e..

0.16,0.23<RETURN>

0.16<RETURN>
0.23<RETURN>

GEAR GEOMETRY PROGRAM.

6.1. CREATING A SET OF DATA.

The following data is required to execute the program.

Assume one is generating the data for the following example, in order to calculate the geometry factors for pitting resistance (I) and bending strength (J).

An existing wheel is to be matched with a new pinion whilst still retaining the existing centre distance of 640 MM.

The 80 MM face width wheel has 89 teeth of module 10, and has been manufactured with an ISO 53 cutter. The measured outside diameter is 1045.0 MM, whilst the measured backlash is 0.23 MM and the helix angle is 30 degrees. Span gauge dimensions indicate that the wheel has no addendum modification, whilst the physical condition is estimated to correspond to an AGMA Quality number of 6.

The proposed new pinion has 21 teeth and is to have an integer outside diameter. The pinion is to be manufactured utilising a nonstandard cutter having the following dimensions; PHIC=20.0, RT = 3 MM, HA = 11.25 MM and HB = 10 MM. The addendum modification coefficient of the pinion is manually calculated to be 0.502, whilst the backlash is to be 0.16 MM. The pinion is to be manufactured to tolerances corresponding to an AGMA Quality number of 8.

Full iteration and different default nominal values are to be used. Before proceeding with the analysis, check whether the wheel has been truncated.

The above information would create the following data set and screen displays as illustrated in the next 20 steps.

Demonstration data set for helical gears.

NP	21	NW	89	MN	10.000	PHIC	20.000
HA(1)	11.250	HA(2)	12.500	HB(1)	10.000	HB(2)	10.000
RT(1)	3.000	RT(2)	3.800	PSIS	30.000	F	80.000
X(1)	.502	X(2)	.000	DLTARO(1)	.000	DLTARO(2)	1.341
BN(1)	0.160	BN(2)	0.230	CNTRS	.000	RTEETH	0
ICODE	1	ROUGH	0	BUTT(1)	0	BUTT(2)	0
CIREF	.050	CUREF	.250	CWREF	.500	CTREF	2.100
QV(1)	8	QV(2)	6				

6.1.1. "TITLE" GEAR SET DESCRIPTION.

An identifying description of the gear set of up to 68 CHARACTERS may be used. Terminate the entry with the <RETURN> key.

Example screen display.(Up to 68 CHARACTERS required)

```
  PLEASE ENTER THE TITLE FOR THIS GEAR SET
TITLE = Demonstration data set for helical gears.<RETURN>
```

GEAR GEOMETRY PROGRAM.

6.1.2. "NP" & "NW" NUMBER OF TEETH ON PINION AND WHEEL.

The gear set can only be one of two possibilities; either it is a pinion and wheel or a rack and pinion.

Example screen display.(A ONE CHARACTER value required)

```
  IS THIS A RACK AND PINION GEAR SET ?
(Y/N) = N<RETURN>
```

If a pinion and wheel is being described, then the number of teeth on the pinion and wheel is requested, otherwise only the number of teeth on the pinion is required. See Section 7. RACKS A SPECIAL CASE for more details on rack and pinion gear sets.

Checking is performed to ensure only integer numbers of teeth are entered and that the number of teeth on the wheel is not less than the number of teeth on the pinion. The range of pinion and wheel teeth is from 1 to 100000, the latter being automatically set if the "wheel" is a rack.

If possible, it is suggested that NW should not be an integer multiple of NP.

A WARNING is generated if the number of pinion teeth is less than 10.

AN ERROR is generated if the number of wheel teeth is less than the number of pinion teeth.

Example screen display.(Two INTEGER values required)

```
  PLEASE ENTER THE TEETH NUMBERS
  NUMBER OF PINION TEETH, NUMBER OF WHEEL TEETH
NP, NW = 21,89<RETURN>
```

6.1.3. "MN" NORMAL METRIC MODULE OF CUTTER.

Generally, first choice normal metric modules of cutters are an integer in the normal plane. Strictly speaking the term metric module refers to the transverse plane.

Cutters with metric modules standardised in the transverse plane are virtually unknown. However, when analysising "imperial gears", particular attention must be paid to the term diametral pitch (DP).

By definition, the diametral pitch is measured in the transverse plane, and cutters such as SYKES, have standard diametral pitches in the TRANSVERSE plane. e.g. a 3.5 DP SYKES cutter corresponds to a normal metric module of 6.28487 MM if the helix angle is 30 degrees.

However, the term diametral pitch is liberally employed when referring to cutters which have been standardised in the normal plane, where as the correct terminology should be the normal diametral pitch (Pn). e.g.. a 3.5 normal DP cutter corresponds to a normal metric module of 7.25714 MM.

As the normal metric module must be entered in MM, the user should be careful that they appreciate, whether the diametral pitch, or the normal diametral pitch, is to be utilised in the manufacturing process.

The normal metric module is entered in MM. This is equivalent to 25.4 / Pn (Normal Diametral Pitch).

A WARNING is generated if the normal metric module of the cutter is less than 1.0 or greater than 50.0.

Example screen display.(One REAL value required)

```
   PLEASE ENTER THE NORMAL METRIC MODULE OF THE CUTTER IN MM.
   NOTE:- NORMAL METRIC MODULE = 25.4 / Pn
          Pn = NORMAL DIAMETRAL PITCH
MN = 10.0<RETURN>
```

USE STANDARD ISO 53 CUTTER.

If a standard ISO 53 cutter (as recommended by AS 2938-1987) is used then the values for the cutter are automatically calculated as follows.

Normal pressure angle	PHIC	= 20.0 Degrees
Addendum of the cutting tool	HA	= 1.25 * MN
Dedendum of the cutting tool	HB	= 1.00 * MN
Radius of the cutting tool	RT	= 0.38 * MN

Note:- If this software has been purchased by a user who would wish to have a cutter other than ISO 53 as the standard, please contact E. Hutton or R. Davey (Phone 042-291282).

However, specified values may be entered.

When entering values for HA & HB, the proceeding comments in regard to the transverse and normal plane are also applicable; e.g. If the "SUNDERLAND" process is to be employed for the generation of double-helical gears, and the cutter has SYKES proportions of "HA"=1.15 and "HB"=0.8 when expressed in terms of unit diametral pitch, then the nominal tooth depth will be 1.95/diametral pitch.

However, when expressed in terms of the normal diametral pitch as required by this program, HA = 1.15 / COS(helix angle) and HB = 0.8/COS(helix angle). If the cutter has a 30 degree helix angle, the nominal tooth depth is 2.2517/normal diametral pitch, which is similar to the 2.25 * MN for an ISO 53 cutter. If a pinion cutter is to be utilised, then specialised gear design knowledge and additional software to GEARGEOM is required. i.e. A SYKES cutter was originally designed as a pinion cutter. However, some manufactures have adapted the proportions to suit their SUNDERLAND gear cutting machines, such that there are "SYKES" rack cutters in existence.

Note:- Differently proportioned cutters may be used on the pinion and wheel. This is especially advantageous when analysing rack and pinion gear sets.

Example screen display.(A ONE CHARACTER value required)

```
  DO YOU WISH TO USE THE STANDARD ISO 53 CUTTER
  AS RECOMMENDED BY THE AUSTRALIAN STANDARD AS2938-1987 ?
(Y/N) = N<RETURN>
```

Note :- Questions 4, 5, 6 and 7 will only be asked if the above answer is N.

6.1.4. "PHIC" NORMAL PRESSURE ANGLE OF CUTTER.

The normal pressure angle of the cutter is entered in the form of Degrees, minutes, seconds. Generally, the normal pressure angle of the cutter will be an integer number of degrees. However, some cutters, such as the SYKES, have pressure angles which have been standardised in the transverse plane. If the pressure angle in the transverse plane is 20 degrees, then the normal pressure angle would be 17 degrees 29 minutes 43 seconds, if the helix angle were 30 degrees. The normal pressure angle would be entered as 17.2943.

A WARNING is generated if the normal pressure angle is less than 14 Degrees 30 Minutes or greater than 25 Degrees.

Example screen display.(One REAL value required)

```
  PLEASE ENTER THE NORMAL PRESSURE ANGLE OF THE CUTTER
  IN DEGREES, MINUTES & SECONDS.
  E.G. A pressure angle of 17 Deg. 29 Min. 43 Sec. would be
entered as 17.2943
PHIC = 20.0000<RETURN>
```

NOTE FOR SPECIAL CUTTERS.

GEARGEOM operates on the premise of the cutting tool being equivalent to a counterpart rack, where the tooth space at the reference pitch line is 0.5 * PI * MN.

GEAR GEOMETRY PROGRAM.

6.1.5. "HA" ADDENDA OF SPECIAL CUTTERS.

Enter the actual addenda of the equivalent counterpart racks to be used. The addendum of a counterpart rack will cut the dedendum of the gear. HA will correspond to the "dedenda" of the actual cutters employed.

A WARNING is generated if the addenda are less than 1.0 * MN or greater than 1.25 * MN.

Example screen display.(Two REAL values required)

```
  PLEASE ENTER THE ADDENDA OF SPECIAL CUTTERS IN MM
  ADDENDUM OF CUTTER FOR PINION, ADDENDUM OF CUTTER FOR WHEEL
HA(1), HA(2) = 11.25,12.5<RETURN>
```

6.1.6. "HB" DEDENDA OF SPECIAL CUTTERS.

Enter the actual dedenda of the equivalent counterpart racks to be used. The dedendum of a counterpart rack will cut the addendum of the gear. HB corresponds to the "addenda" of the basic racks and hence will be less than that of the actual cutters employed. (Refer Figs. B1; B2; B3; B5; B6; B7 & B8 of AGMA 218.01)

A WARNING is generated if the dedenda are less than 0.8 * MN or greater than 1.0 * MN.

Example screen display.(Two REAL values required)

```
  PLEASE ENTER THE DEDENDA OF SPECIAL CUTTERS IN MM
  DEDENDUM OF CUTTER FOR PINION, DEDENDUM OF CUTTER FOR WHEEL
HB(1), HB(2) = 10.0,10.0<RETURN>
```

6.1.7. "RT" TIP RADII OF SPECIAL CUTTERS.

Enter the actual tip radii of the equivalent counterpart racks to be used. RT will correspond to that of the actual cutters employed.

A WARNING is generated if the tip radius is less than 0.1 * MN.

An ERROR is generated if the tip radius is greater than
(0.25 * PI * MN * COS(PHIC) - HA * SIN(PHIC)) /
(1.0 - SIN(PHIC))

Example screen display.(Two REAL values required)

```
  PLEASE ENTER THE TIP RADII OF SPECIAL CUTTERS IN MM
  TIP RADIUS OF CUTTER FOR PINION, TIP RADIUS OF CUTTER FOR
WHEEL
RT(1), RT(2) = 3.0,3.8<RETURN>
```

GEAR GEOMETRY PROGRAM.

6.1.8. "PSIS" HELIX ANGLE.

A gear set can only be a spur gear set or a helical gear set.

Example screen display.(A ONE CHARACTER value required)

IS THIS A SPUR GEAR SET ?
(Y/N) = N<RETURN>

If a helical gear set is being described then the helix angle at the reference pitch circle diameter is requested in the form of Degrees, Minutes, Seconds. e.g. if the helix angle is 17 Degrees 29 Minutes 43 Seconds then the entry 17.2943 is used.

A WARNING is generated if the helix angle is less than 5 Degrees or greater than 30 Degrees.

An ERROR is generated if the helix angle is greater than 45 Degrees or less than or equal to 0.

Example screen display.(One REAL value required)

PLEASE ENTER THE HELIX ANGLE IN DEGREES, MINUTES & SECONDS.
E.G. A helix angle of 17 Deg. 29 Min. 43 Sec. would be entered as 17.2943
PSIS = 30.0000<RETURN>

Note:- The helix angle will only be asked for if the above answer is N.

6.1.9. "F" FACE WIDTH OF THE NARROWEST MEMBER OF THE GEAR SET.

The face width of the narrowest member of the gear set is entered in millimetres.

A WARNING is generated for spur gears if the face width is less than 5 * MN or greater than 10 * MN, and for helical gears the WARNING is generated if the face width is less than PI * MN / SIN(PSIS) or greater than 20 * MN.
Note :- For herringbone and double helical gears F is the width of one helix only.

Example screen display.(One REAL value required)

PLEASE ENTER THE FACE WIDTH OF THE NARROWEST GEAR IN MM
F = 80.0<RETURN>

GEAR GEOMETRY PROGRAM.

6.1.10. "X" ADDENDUM MODIFICATION COEFFICIENTS.

The addendum modification coefficients for the pinion and wheel are entered.

A WARNING is generated if either of the addendum modification coefficients are less than -0.5 or greater than 0.5.

A positive coefficient will generally produce an enlargement of the gear's dimensions, whilst a negative coefficient will generally produce a reduction in the gear's dimensions.

Example screen display.(Two REAL values required)

```
  PLEASE ENTER THE ADDENDUM MODIFICATION COEFFICIENT
  ADDENDUM COEF. PINION, ADDENDUM COEF. WHEEL
X(1), X(2) = 0.502,0.0<RETURN>
```

Verification is required because the addendum modification coefficient for the pinion exceeds a warning limit .

Example screen display.(One REAL value required)

```
WARNING
  ENTERED ADDENDUM MODIFICATION COEFFICIENT FOR PINION = .502
  CUSTOMARY ADDENDUM MODIFICATION COEFFICIENT
  BETWEEN  -.500 TO .500
  PLEASE VERIFY ADDENDUM COEF. FOR THE PINION
X(1) = 0.502<RETURN>
```

6.1.11. "DLTARO" TRUNCATION APPLIED (TOPPING).

Truncation applied to either of the gears is entered in millimetres. Positive truncation will decrease the outside diameter of the gear, whilst negative truncation will increase the outside diameter of the gear. If negative truncation is used, special attention must be directed to the cutter to ensure that there is sufficient clearance for manufacture. The actual dedendum of the cutter must be substantially greater than that of the basic rack. i.e. for an ISO 53 cutter
HB > 1.00 * MN. + DLTARO
A WARNING is generated if the Truncation applied is less than 0.0 or greater than 0.5 * MN.

Example screen display.(Two REAL values required)

```
  PLEASE ENTER THE TRUNCATION APPLIED IN MM
  TRUNCATION OF PINION, TRUNCATION OF WHEEL
DLTARO(1), DLTARO(2) = 0.0,1.341<RETURN>
```

GEAR GEOMETRY PROGRAM.

6.1.12. "BN" NORMAL BACKLASH APPLIED.

The backlash applied to either the pinion and/or wheel in the normal plane is entered in millimetres.

A WARNING is generated if the backlash is less than 0.0 or greater than 0.1 * MN.

Example screen display.(Two REAL values required)

```
  PLEASE ENTER THE NORMAL BACKLASH APPLIED IN MM
  BACKLASH OF PINION, BACKLASH OF WHEEL
BN(1), BN(2) = 0.16,0.23<RETURN>
```

6.1.13. "ICODE" TYPE OF ITERATION.

The type of iteration performed to solve the iterative steps in the analysis of the entered gear set is controlled by this variable. A "0" will only iterate three times towards the answer, whilst a "1" will continue iterations until the absolute difference between two consecutive steps is less than 0.00000001. The only values accepted are either "0" or "1". However it is suggested that if the sum of the addendum modification coefficients is not zero, then ICODE = 1 should be used, such that the operating transverse pressure angle may be accurately calculated. If a problem occurs when using FULL iteration whilst analysing racks, then use CODE iteration.

Example screen display.(One INTEGER value required)

```
  ITERATION TYPE REQUIRED
  0 = CODE ITERATION
  1 = FULL ITERATION
  N.B. FULL ITERATION TO 0.00000001
ICODE = 1<RETURN>
```

6.1.14. "ROUGH" ACCURATE SPUR GEARS.

The variation in the base pitch between the pinion and wheel, determines whether or not load sharing exists in steel spur gears. Base pitch errors in excess of those shown in Table 2 of AGMA 218.01 render spur gears to be classified as inaccurate. Inaccurate spur gears are analysed with the load applied at the tooth tip, whist accurate spur gears are analysed with the load applied at the highest point of single tooth contact (HPSTC).

Note:- This question will not be asked in the current demonstration, but had this question been asked then the following screen would have been displayed.(One INTEGER value required)

```
  ARE THE SPUR GEARS ACCURATELY MACHINED ?(0=YES, 1=NO )
ROUGH = 0<RETURN>
```

GEAR GEOMETRY PROGRAM.

6.1.15. "BUTT" BUTTRESSING OF THE PINION AND WHEEL.

The buttressing of a pinion is indicated with a "1" or negated with a "0". In accordance with clause 6.3.2.3 of AGMA 218.01, the helical factor (CH) may be increased by 10 percent where full buttressing exists. When the face width of the pinion exceeds that of the wheel by at lease two standard addenda (i.e. 2.0 * MN for an ISO 53 cutter) and is so positioned that at least one standard addendum projects beyond each extremity of the face width of the wheel, then the pinion is said to be fully buttressed (see Fig 5 AGMA 218.01). This increased value of CH is only applicable to conventional helical gears.

If a helical gear set is being described, and
F > PI * MN / SIN(PSIS) then these questions are asked.

Example screen display.(One INTEGER value required)

```
   IS THE PINION BUTTRESSED ? (0=NO, 1=YES)
BUTT(1) = 0<RETURN>
```

Example screen display.(One INTEGER value required)

```
   IS THE WHEEL BUTTRESSED ? (0=NO, 1=YES)
BUTT(2) = 0<RETURN>
```

CHANGING DEFAULT NOMINAL VALUES

Due to the universal nature of the program, gear sets can be analysed that are outside the bounds of "orthodox" gear design practice. The program automatically identifies, and suggests solutions to four of these potential errors. Namely that of tip to root fillet interference, insufficient bottom clearance, insufficient top land width and non conjugate action. The default values for these four checks can be changed to allow a particular gear set to be analysed. However, the user is warned that the nominal default values have been set, based on traditional gear design practice and they should not be changed without a thorough understanding of the ramifications.

Example screen display.(A ONE CHARACTER value required)

```
   DO YOU WISH TO ALTER THE DEFAULT NOMINAL VALUES FOR ?
   INVOLUTE CLEARANCE COEFFICIENT      = 0.05
   ROOT CLEARANCE COEFFICIENT          = 0.25
   TOP LAND WIDTH COEFFICIENT          = 0.40
   TOTAL CONTACT RATIO     SPUR        = 1.30
                           HELICAL     = 1.50
(Y/N) = Y<RETURN>
```

6.1.16. "CIREF" MINIMUM INVOLUTE CLEARANCE COEFFICIENT.

Nominal minimum value = 0.05

The involute clearance coefficient ensures that the tip of one gear meshes with the involute of the mating gear and not the trochoidal fillet. The involute clearance coefficient is influenced by applied truncation, addendum modification coefficients and backlash.

A WARNING is generated if the nominated involute clearance coefficient is less than 0.04 or greater than 0.10.

Example screen display.(One REAL value required)

```
  PLEASE ENTER THE MINIMUM INVOLUTE CLEARANCE COEFFICIENT
  CUSTOMARY NOMINAL VALUE = 0.050
CIREF = 0.05<RETURN>
```

6.1.17. "CUREF" MINIMUM ROOT CLEARANCE COEFFICIENT.

Nominal minimum value = 0.25

The root clearance coefficient controls the clearance between the tip of one gear and the root of the mating gear. The root clearance coefficient is influenced by applied truncation, addendum modification coefficients and backlash.

A WARNING is generated if the root nominated clearance coefficient is less than 0.20 or greater than 0.30.

Example screen display.(One REAL value required)

```
  PLEASE ENTER THE MINIMUM ROOT CLEARANCE COEFFICIENT
  CUSTOMARY NOMINAL VALUE = 0.25
CUREF = 0.25<RETURN>
```

6.1.18. "CWREF" MINIMUM TOP LAND WIDTH COEFFICIENT.

Nominal minimum value = 0.40

The top land width coefficient controls the width of the tooth at the tip to ensure that there is sufficient thickness; (certain combinations may remove the tip altogether leaving a "triangular" shaped tooth). For gears that are to be surface heat treated, the tip thickness should be increased in conformity with good engineering practice. The top land width coefficient is influenced by applied truncation, addendum modification coefficients and backlash.

GEAR GEOMETRY PROGRAM.

A WARNING is generated if the nominated top land width coefficient is less than 0.25 or greater than 0.5.
Example screen display.(One REAL value required)

PLEASE ENTER THE MINIMUM TOP LAND WIDTH COEFFICIENT
CUSTOMARY NOMINAL VALUE = 0.40
CWREF = 0.50<RETURN>

6.1.19. "CTREF" MINIMUM TOTAL CONTACT RATIO.

Nominal minimum value equals 1.3 for spur gears and 1.5 for helical gears.

The conjugate operation of a the gear set is only maintained if the value calculated for MT is greater than unity.

A WARNING is generated for spur gears if the nominated total contact ratio is less than 1.2 or greater than 1.7.

A WARNING is generated for helical gears if the total contact ratio is less than 1.4 or greater than 4.0.

Example screen display.(One REAL value required)

PLEASE ENTER THE MINIMUM TOTAL CONTACT RATIO.
CUSTOMARY NOMINAL VALUE SPUR = 1.3, HELICAL = 1.5
CTREF = 2.1<RETURN>

6.1.20. "QV" QUALITY NUMBERS.

Nominal value = 6.

The quality numbers of the manufactured gears are required for display in the specification of the gear set, and are also necessary for the determination of the allowable transmitted power rating as per AGMA 218.01, should IRATE be used to complement GEARGEOM.

A WARNING is generated if the Quality number is less than 6 or greater than 10.

Example screen display.(Two INTEGER values required)
PLEASE ENTER THE QUALITY NUMBERS FOR THE PINION AND WHEEL.
NOMINAL VALUE = 6
QV(1),QV(2) = 8,6<RETURN>

GEAR GEOMETRY PROGRAM

6.2 RETRIEVING A STORED SET OF DATA.

Selecting a "2" from the INITIAL SET OF DATA menu enters the RETRIEVE AN EXISTING SET OF DATA mode.

Example screen display. (Up to 45 CHARACTERS required)

```
****************************************************************
*                                                              *
*                                                              *
*                                                              *
* GEAR GEOMETRY PROGRAM  MODE :- RETRIEVE AN EXISTING SET OF DATA *
*                                                              *
*                                                              *
*                                                              *
****************************************************************

  PLEASE ENTER FILE NAME OF EXISTING SET OF DATA
NAME = EXAMPLE1.DAT<RETURN>
```

NOTE:- The file name can contain a drive identifier, subdirectories names and a file name. This allows for saving or retrieving files from other than the current drive and directory. Refer to your DOS manual for more information about file names and directory paths.

If the specified file was opened, then the data is read from the file and control is passed to the SELECTED DATA menu via the EDIT mode, otherwise the following warning is displayed and control returned to the INITIAL SET OF DATA menu.

WARNING ==>   FILE NAME DOESN'T EXIST

6.3 SAVING THE SET OF DATA.

Selecting a "3" (SAVE) from the SELECTED SET OF DATA menu enters the SAVE DATA FILE mode.

Example screen display. (Max 45 "DOS FILE" CHARACTERS allowed)

```
****************************************************************
*                                                              *
*  GEAR GEOMETRY PROGRAM   MODE :- SAVE DATA FILE              *
*                                                              *
*  FILE NAME = ...none...                                      *
*                                                              *
****************************************************************

  PLEASE ENTER FILE NAME TO SAVE DATA SET
FILE NAME = DEMO1.DAT<RETURN>
```

GEAR GEOMETRY PROGRAM

The file specification can contain a drive identifier, subdirectories names and a file name. (Refer Section 6.2) This allows for saving or retrieving files from other than the current drive and directory. If the specified file can be accessed, then the data is written to disk, and the following screen is displayed. Otherwise an error screen is displayed informing the user that the file was not saved. After pressing RETURN to continue, the user is returned to the SELECTED DATA menu and the file name will be displayed if the save was successful.

```
***************************************************************************
*                                                                         *
*  GEAR GEOMETRY PROGRAM    MODE :- SAVE DATA FILE                        *
*                                                                         *
*  FILE NAME = EXAMPLE1.DAT                                               *
*                                                                         *
*  DATA FILE HAS BEEN SAVED                                               *
*                                                                         *
***************************************************************************
```

Press RETURN to continue

6.4 ANALYSING THE SET OF DATA.

Selecting a "1" (ANALYSE) from the SELECTED DATA menu enters the analysing data mode.

Example screen display. (One INTEGER value required)

```
***************************************************************************
*                                                                         *
*  CALCULATION OF THE GEOMETRY FACTORS FOR PITTING RESISTANCE AND         *
*  BENDING STRENGTH FOR EXTERNAL SPUR AND HELICAL INVOLUTE GEAR           *
*  TEETH WITHOUT PROTUBERANCE                                             *
*                                                                         *
*  GEAR GEOMETRY PROGRAM    MODE :- ANALYSE GEAR SET                      *
*                                                                         *
*  FILE NAME = EXAMPLE1.DAT                                               *
*                                                                         *
*  PLEASE SELECT DESTINATION OF CALCULATIONS                              *
*                                                                         *
*  1 = SCREEN                                                             *
*  2 = PRINTER                                                            *
*  3 = FILE                                                               *
*                                                                         *
***************************************************************************
```

SELECT OPTION ==> 1<RETURN>

The results of the analysis can be displayed on the screen, sent to the printer or stored on to a diskette or hard disk. A drive, directories names and file name may be specified. (Refer Section 6.3)

NOTE :- The destination of the calculations can only be directed to one location at a time. It is suggested that the screen be used until a printed output is required. If a hard copy of a particular screen is required then the print screen function can be utilised (While holding down a <SHIFT> key press the <PrtSc> key), When you require a printed copy of the analysis, analyse the set of data again and direct the output to your printer.

6.5 EDITING THE GEAR SET DATA.

After creating a set of data or retrieving an existing set of data or selecting option "2" (EDIT) from the SELECTED SET OF DATA menu, the following screen is displayed. Verification and/or correction of the current gear set data is then possible.

NOTE:- A rack and pinion type gear set can not be altered to a pinion and wheel type gear set, or vice-versa, in the EDIT mode. Similarly, a helical gear set can't be altered to a spur gear set, or vice-versa, in the EDIT mode. To effect these types of changes, you will have to CREATE a new set of data.

```
*****************************************************************
*   Description                            Variable  Pinion  Wheel *
*****************************************************************
* 1 TITLE Demonstration data set for helical gears.                *
* 2 Number of teeth                          NP, NW :    21      89 *
* 3 Normal metric module of cutter   (MM)       MN :  10.000        *
* 4 Normal pressure angle of cutter            PHIC :  20 Deg  0  0  *
* 5 Addenda of cutters               (MM)       HA :  11.250  12.500 *
* 6 Dedenda of cutters               (MM)       HB :  10.000  10.000 *
* 7 Tip radii of cutters             (MM)       RT :   3.000   3.800 *
* 8 Helix angle at reference PCD               PSIS :  30 Deg  0  0  *
* 9 Face width of narrowest gear     (MM)        F :  80.000        *
*10 Addendum modification coefficients           X :    .502    .000 *
*11 Truncations applied              (MM) DLTARO :    .000   1.341 *
*12 Normal backlash on each gear     (MM)       BN :    .160    .230 *
*13 Code iteration          (0=Yes,1=No)  ICODE :       1           *
*14 Accurate spur gear      (0=Yes,1=No)  ROUGH :       0           *
*15 Buttressing             (1=Yes,0=No)   BUTT :       0       0 *
*16 Minimum involute clearance coeff.     CIREF :    .050           *
*17 Minimum root clearance coefficient    CUREF :    .250           *
*18 Minimum top land width coefficient    CWREF :    .500           *
*19 Minimum for total contact ratio       CTREF :   2.100           *
*20 Quality numbers                          QV :       8       6 *
*****************************************************************
Enter a NUMBER (1 to 20) to ALTER or 0 when FINISHED
```

GEAR GEOMETRY PROGRAM

To alter any data, type the number (1 to 20) corresponding to the variable to be altered and press the ENTER key, then answer the question/s. The above screen will be redisplayed with the altered data. Continue until you have finished editing, then enter a zero (0) and press the RETURN key.

When you are familiar with the operation of GEARGEOM, it may be quicker to retrieve a stored data file (Section 6.2) and use the EDIT mode, rather than create a new set of data as detailed in Section 6.1. Please refer to Section 6.3.

From an inspection of the output of the analysis for the set of data generated in Section 6.1, STEP 6 indicates that R0 (Tip Radius) for the pinion is 136.264 MM. In order to obtain an integer outside diameter, one option is to truncate the pinion by 0.264 MM.

Selection of a "2" (EDIT) from the SELECTED SET OF DATA menu enters the EDIT mode . Selection of an "11" enables the truncation for the pinion and wheel to be changed from their existing numerical values.

Example screen display. (Two REAL values required)

```
   CURRENT TRUNCATION APPLIED IN MM
   FOR PINION AND WHEEL
   DLTARO(1), DLTARO(2) = 0.000, 1.341
   PLEASE ENTER THE TRUNCATION APPLIED IN MM
   TRUNCATION OF PINION, TRUNCATION OF WHEEL
DLTARO(1),DLTARO(2) = 0.264,1.341<RETURN>
```

The EDIT screen is redisplayed, verifying that the truncation has been altered. Selection of a "0" will leave the EDIT mode.

Analysing the gear set again, will produce the following output.

6.6. PRINTED OUTPUT FOR HELICAL EXAMPLE.

GEAR SET 1 FULL ITERATION

Demonstration data set for helical gears.

GEOMETRY FACTOR-CONVENTIONAL HELICAL GEARS-AS2938-1987.

STP	NAME	PINION	WHEEL	UNIT	DESCRIPTION
I	NP,NW	21	89		NUMBER OF TEETH
N	MN	10.000	10.000	MM	NORMAL METRIC MODULE
P	PHIC	20 00'00"	20 00'00"	DEG	NORMAL PRESSURE ANGLE
U	HA	11.250	12.500	MM	STANDARD ADDENDUM OF TOOL
T	HB	10.000	10.000	MM	STANDARD DEDENDUM OF TOOL
	RT	3.000	3.800	MM	TIP RADIUS OF CUTTING TOOL
D	PSIS	30 00'00"	30 00'00"	DEG	HELIX ANGLE AT REF PCH DIA.
A	F	80.000	80.000	MM	NET FACE WIDTH
T	X	.502	.000		ADDENDUM MODIF. COEFF
A	DLTARO	.264	1.341	MM	TRUNCATION APPLIED
	BN	.160	.230	MM	BACKLASH APPLIED
	BUTT	0	0		BUTTRESSING (0=NO, 1=YES)
	QV	8	6		AGMA 390 QUALITY NUMBER

INVOLUTE GEAR MATHEMATICS

GEOMETRY FACTOR-CONVENTIONAL HELICAL GEARS-AS2938-1987.

STP	NAME	PINION	WHEEL	UNIT	DESCRIPTION
6A	RS	121.244	513.842	MM	REFERENCE PITCH RADII
6A	SMALLC	1.974	2.500	MM	CUTTER TIP RADII CONSTANT
6A	RU	101.069	434.205	MM	UNDERCUT RADII
6A	RR	114.794	501.026	MM	ROOT RADII
6A		NO	NO		UNDERCUT (YES=RU>RR)
6A	BIGRF	117.252	504.124	MM	RADII TO TOP OF TROCHOID
6	RO	136.000	522.501	MM	ORIGINAL TIP RADII
6A	ROM	141.043	523.597	MM	MAX ALLOW TIP RADII
6A	CI	.299	.081	MM	INVOLUTE CLEAR. COEFFICIENT
6A		NO	NO		TIP INTERFER. (YES=CI< .05)
6A	CU	.271	.297	MM	BOTTOM CLEAR. COEFFICIENTS
6A		NO	NO		ROOT INTERFER.(YES=CU< .25)
6A	TST	22.173	17.872	MM	REF TRANSV. ARC TOOTH WIDTH
6A	PSIO	32 55'40"	30 24'59"	DEG	TIP HELIX ANGLES
6A	PHIO	34 43'41"	24 57'31"	DEG	TIP TRANSV. PRESSURE ANGLES
6A	TOT	7.293	10.436	MM	TIP TRANSV. ARC TOOTH WIDTH
6A	BETAH	1 04'57"	0 25'32"	DEG	HALF TIP TRANSVERSE ANGLES
6A	TO	6.121	8.999	MM	NORMAL TOP LAND WIDTHS
6A	CW	.612	.900		TOP LAND WIDTH COEFFICIENTS
6A		NO	NO		LAND TOO SMALL (YES=CW<.50)
6A	RMID	126.626	513.312	MM	RADII TO MID POINT INVOLUTE
6A	ANC	9.544	9.234	MM	CHORDAL HEIGHTS @ RMID
6A	TNC	15.324	15.849	MM	CHORDAL WIDTHS @ RMID
6A	SPAN	111.021	121.840	MM	SPAN DIMENSION
6A	STEETH	4.000	4.000		OVER NUMBER OF TEETH
6A	FMIN	52.163	57.246	MM	MIN FACE WIDTH FOR SPAN DIM
6A	EPSLON	.814	.369		SLIDE/ROLL RATIOS
6A	MF	1.273	1.273		FACE CONTACT RATIO
6A	MP	1.180	1.180		TRANSVERSE CONTACT RATIO
6A	MT	2.453	2.453		TOTAL CONTACT RATIO
6A		NO	NO		NON-CONJUGATE ACTN(MT<=2.1)

GEAR GEOMETRY PROGRAM

GEAR SET 1 FULL ITERATION
DEMONSTRATION data set for helical gears.
GEOMETRY FACTOR-CONVENTIONAL HELICAL GEARS-AS2938-1987.

STP	NAME	PINION	WHEEL	UNIT	DESCRIPTION
1	PHIS	22 47'45"	22 47'45"	DEG	TRANSVERSE PRESSURE ANGLE
2	PHIT	23 49'16"	23 49'16"	DEG	OPERATING TRANSV. PRESS ANG
3	C	640.000	640.000	MM	OPERATING CENTRE DISTANCE
4	RB	111.773	473.706	MM	BASE RADII
5	R	122.182	517.818	MM	OPERATING PITCH RADII
6	R0	136.000	522.501	MM	TIP RADII
7	MF	1.273	1.273		FACE CONTACT RATIO
8	ZB	11.339	11.339	MM	LENGTH OF APPROACH PATH
8	ZA	28.130	28.130	MM	LENGTH OF RECESS PATH
9	Z	39.469	39.469	MM	LENGTH OF LINE OF ACTION
10	ZC	-10.420	-10.420	MM	DIST STRESS TO PITCH POINTS
11	CX	1.151	1.151		CONTACT HEIGHT FACTOR
13	PSIB	28 01'28"	28 01'28"	DEG	BASE HELIX ANGLE
14	CPSI	1.000	1.000		HELICAL OVERLAP FACTOR
15	CC	.149	.149		CURVATURE FACTOR
16	LMIN	103.452	103.452	MM	MIN LGH OF LINES OF CONTACT
17	MNN	.773	.773		LOAD SHARING RATIO
18	I	.222	.222		GEOMETRY FACTOR - PITTING
19	PSI	30 11'30"	30 11'30"	DEG	OPERAT PITCH DIA HELIX ANG
20	PHIN	20 53'13"	20 53'13"	DEG	OPERATING NORMAL PRESS ANG
21	NE	32.520	137.824		VIRTUAL NO. OF SPUR TEETH
21	ROE	177.357	697.777	MM	VIRTUAL SPUR TIP RADII
22	BETA	.533	.382	RAD	BASE & TIP RADII INCLD ANGS
22	PHIL	.515	.375	RAD	LOAD ANGLES AT TIP
23	LAMBAI	.262	.262	RAD	LAMBDA - INITIAL VALUE
24	K1	.345	.902	MM	CONSTANT - K1
24	K2	1.405	1.506	MM	CONSTANT - K2
24	K3	47.133	76.433		CONSTANT - K3
25	K4	1.191	1.234		VARIABLE - K4
25	K5	1.166	1.218	RAD	VARIABLE - K5
25	K6	1.144	1.200	RAD	VARIABLE - K6
25	K7	1.144	1.200	RAD	VARIABLE - K7
25	K8	.133	.056	RAD	VARIABLE - K8
25	K9	1.133	2.870	MM	VARIABLE - K9
26	TE	21.447	21.934	MM	INSCRIBED PARABOLA - WIDTHS
26	HE	17.726	16.826	MM	INSCRIBED PARABOLA - HEIGHT
27	LAMBDA	.294	.315	RAD	LAMBDA - FINAL VALUES
28	CH	1.474	1.474		HELICAL FACTOR
29	KPSI	.749	.749		HELIX ANGLE FACTOR
30	Y	.616	.604		TOOTH FORM FACTORS
31	B	7.388	16.792	MM	OPERATING DEDENDA
31	RF	3.115	4.040	MM	MINIMUM ROOT FILLET RADII
31	H	.173	.173		DOLAN - BROGHAMER FACTOR -H
31	L	.143	.143		DOLAN - BROGHAMER FACTOR -L
31	M	.459	.459		DOLAN - BROGHAMER FACTOR -M
31	KF	1.611	1.611		STRESS CONCENTRATION FACTOR
32	J	.495	.485		GEOMETRY FACTORS - STRENGTH

GEAR GEOMETRY PROGRAM

GEAR SET 1 FULL ITERATION
DEMONSTRATION data set for helical gears.
THE FOLLOWING TABULATION SHOULD BE INCLUDED ON ENGINEERING DRAWINGS.

CONVENTIONAL HELICAL GEARS TO AS2938-1987. MODIFIED ADDENDA

HOB OR COUNTERPART RACK DETAILS	UNIT	PINION	WHEEL
NORMAL MODULE	MM	10.000	10.000
NORMAL PRESSURE ANGLE	DMS	20 00'00"	20 00'00"
TIP RADII	MM	3.000	3.800
STANDARD ADDENDA	MM	11.250	12.500
STANDARD DEDENDA	MM	10.000	10.000

DESIGN DETAILS	UNIT	PINION	WHEEL
NUMBER OF TEETH		21	89
HELIX ANGLE AT REFERENCE PCD	DMS	30 00'00"	30 00'00"
ADDENDUM MODIFICATION COEFFICIENTS		.502	.000
RADIAL TOOTH TRUNCATIONS (TOPPING)	MM	.264	1.341
OPERATING NORMAL PRESSURE ANGLE	DMS	20 53'13"	20 53'13"
REFERENCE PITCH CIRCLE DIAMETERS	MM	242.487	1027.683
OPERATING PITCH CIRCLE DIAMETERS	MM	244.364	1035.636
OPERATING CENTRE DISTANCE	MM	640.000	640.000
OPERATING ADDENDA	MM	13.818	4.683
OPERATING DEDENDA	MM	7.388	16.792
ACTUAL DEPTH OF TEETH	MM	21.206	21.475
NOMINAL DEPTH OF TEETH	MM	21.250	22.500
INVOLUTE CLEARANCES	MM	2.987	.806
BOTTOM CLEARANCES	MM	2.705	2.975
SLIDE/ROLL RATIOS		.814	.369
TOTAL CONTACT RATIO		2.453	2.453
GEOMETRY FACTOR - PITTING		.222	.222
GEOMETRY FACTORS - STRENGTH		.495	.485

MACHINING DETAILS	UNIT	PINION	WHEEL
QUALITY NUMBERS (AGMA 390)		8	6
NORMAL THINNING OF TEETH(BACKLASH)	MM	.160	.230
OUTSIDE DIAMETERS	MM	271.999	1045.001
ROOT DIA. INCLUDING BACKLASH	MM	229.588	1002.052
RADII TO MID POINT OF INVOLUTE	MM	126.626	513.312
CHORDIAL HEIGHTS @ RMID	MM	9.544	9.234
CHORDIAL WIDTHS @ RMID	MM	15.324	15.849
SPAN GAUGE DIMENSIONS	MM	111.021	121.840
NUMBER OF TEETH SPANNED		4	4
TOP LAND WIDTHS	MM	6.121	8.999

NOTE: CHORDAL & SPAN DIMENSIONS HAVE BACKLASH INCORPORATED

DESIGNER: ____________________ DATE : __________

GEAR GEOMETRY PROGRAM.

7. RACKS - A SPECIAL CASE.

A rack is a toothed member which may be regarded as a portion of a wheel of infinite radius.

One common error in the design of a rack and pinion, is that of not investigating involute clearances. If the number of wheel teeth is set to 100,000, GEARGEOM will analyse a rack and pinion, to a degree of accuracy commensurate with traditional equations unique to a rack and pinion.

The advantage of this technique, is the ability to mathematically model the rack to suit the manufacturing process, noting that X(2) always equals zero.

However, to obtain meaningful output, two additional variables must be defined. Firstly, the actual number of teeth to be cut in the rack, (RTEETH) is entered as data. The program will automatically set the number of wheel teeth as 100,000 for the purpose of analysis.

Secondly, the "centre distance" between the rack and pinion (CNTRS) is also entered as data. Whilst in the orthodox sense, centre distance can not be defined as other than infinity, contemplate the example of a hydraulic actuator. Traditionally, the rack is part of the hydraulic piston, and in this case, the centre distance may be defined as, the distance between the centre of the pinion and the centre of the hydraulic cylinder.

7.1 AN EXAMPLE.

Consider the design of a spur rack to suit a hydraulic actuator. The pinion has an outside diameter of 376 MM, 16 teeth, an addendum modification of 8 MM, backlash of 0.132 MM and is 235 MM wide. The pinion has been manufactured with a 20 normal metric module ISO 53 cutter, to tolerances corresponding to an AGMA quality number of 6.

The eleven teeth of the rack are to be cut on a milling machine, to tolerances corresponding to an AGMA quality number of 5, and the backlash is to be 0.5 MM. The maximum fillet radius that can be milled is 4 MM, whilst the depth of the rack teeth are to be minimised, to aid manufacturing cost. The distance between the centre line of the pinion and the centre line of the hydraulic cylinder is 200 MM.

As a first attempt solution to the problem, it would appear reasonable to assume ISO 53 rack proportions and nominal default values. The following data file is created and entered, from which GEARGEOM indicates the following;

GEAR GEOMETRY PROGRAM.

Demonstration data set for a rack and pinion gear set

NP	16	NW	100000	MN	20.000	PHIC	20.000
HA(1)	25.000	HA(2)	25.000	HB(1)	20.000	HB(2)	20.000
RT(1)	7.600	RT(2)	4.000	PSIS	.000	F	235.000
X(1)	.400	X(2)	.000	DLTARO(1)	.000	DLTARO(2)	.000
BN(1)	.132	BN(2)	.500	CNTRS	200.000	RTEETH	11
ICODE	1	ROUGH	0	BUTT(1)	0	BUTT(2)	0
CIREF	.050	CUREF	.250	CWREF	.400	CTREF	1.300
QV(1)	6	QV(2)	5				

NAME	PINION	WHEEL	DESCRIPTION
DLTAR3	.000	2.131	TRUNCATION APPLIED FOR CI
DLTAR4	.000	.000	TRUNCATION APPLIED FOR CU
DLTAR5	.000	.000	TRUNCATION APPLIED FOR LAND

TO RECTIFY RADIAL TOOTH TRUNCATION IS NECESSARY
MINIMUM RADIAL TOOTH TRUNCATION OF PINION DLTARO(1) = .000
MINIMUM RADIAL TOOTH TRUNCATION OF WHEEL DLTARO(2) = 2.131
THESE MODIFICATIONS YIELD THE FOLLOWING RELATIONSHIPS.

FACE CONTACT RATIO	MF =	.000
TRANSVERSE CONTACT RATIO	MP =	1.473
TOTAL CONTACT RATIO	MT =	1.473
PINION INVOLUTE CLEARANCE COEFFICIENT	CI(1) =	.588
WHEEL INVOLUTE CLEARANCE COEFFICIENT	CI(2) =	.050
PINION BOTTOM CLEARANCE COEFFICIENT	CU(1) =	.366
WHEEL BOTTOM CLEARANCE COEFFICIENT	CU(2) =	.284
PINION TIP WIDTH COEFFICIENT	CW(1) =	.453
WHEEL TIP WIDTH COEFFICIENT	CW(2) =	.895

The suggested wheel truncation of 2.132 MM can be reduced to 2.000 MM or alternatively, HB can be reduced from 20.0 to 18.0 MM. To enable the analysis to proceed, CIREF will need to be reduced from 0.05 to say 0.04.

The proceding tabulation indicates that there is excessive involute and bottom clearance on the pinion, ie. the rack teeth can be reduced in height, which is in keeping with minimising the amount of machining.

The depth of the rack teeth is controlled by the variable HA, plus an additional amount due to the backlash of
0.5 * BN / TAN(PHIC). One solution is to reduce HA by 1 MM eg.
HA = 25.0 - 1.0 - 0.5 * 0.5 / TAN(20.0). ie. HA = 23.313
To enable the analysis to proceed, CUREF will need to be reduced from 0.25 to 0.20.

The modifications to HA and HB yield an integer tooth height as shown in the following tabulation.

GEAR GEOMETRY PROGRAM.

When a rack is represented by a wheel of 100000 teeth, an appreciation of the degree of accuracy of GEARGEOM can be asertained by comparing the computed values with those obtained from Euclidean geometry. For example, the chordal heights and widths obtained from GEARGEOM are 19.683 MM and 32.141 MM respectively. whilst the actual values are 19.684 MM and 32.110 MM respectively.

Demonstration data set for a rack and pinion gear set.

NP	16	NW	100000	MN	20.000	PHIC	20.000
HA(1)	25.000	HA(2)	23.313	HB(1)	20.000	HB(2)	18.000
RT(1)	7.600	RT(2)	4.000	PSIS	.000	F	235.000
X(1)	.400	X(2)	.000	DLTARO(1)	.000	DLTARO(2)	.000
BN(1)	.132	BN(2)	.500	CNTRS	200.000	RTEETH	11
ICODE	1	ROUGH	0	BUTT(1)	0	BUTT(2)	0
CIREF	.040	CUREF	.200	CWREF	.400	CTREF	1.300
QV(1)	6	QV(2)	5				

Note:- The calculations for the span dimensions of the rack are automatically negated.

7.2. PRINTED OUTPUT FOR RACK EXAMPLE.

GEAR SET 1 FULL ITERATION
Demonstration data set for rack and pinion gear set.
GEOMETRY FACTOR - ACCURATE SPUR GEARS - AS2938-1987.

STP	NAME	PINION	WHEEL	UNIT	DESCRIPTION
I	NP,NW	16	100000		NUMBER OF TEETH
N	MN	20.000	20.000	MM	NORMAL METRIC MODULE
P	PHIC	20 00'00"	20 00'00"	DEG	NORMAL PRESSURE ANGLE
U	HA	25.000	23.313	MM	STANDARD ADDENDUM OF TOOL
T	HB	20.000	18.000	MM	STANDARD DEDENDUM OF TOOL
	RT	7.600	4.000	MM	TIP RADIUS OF CUTTING TOOL
D	PSIS	0 00'00"	0 00'00"	DEG	HELIX ANGLE AT REF PCH DIA.
A	F	235.000	235.000	MM	NET FACE WIDTH
T	X	.400	.000		ADDENDUM MODIF. COEFF
A	DLTARO	.000	.000	MM	TRUNCATION APPLIED
	BN	.132	.500	MM	BACKLASH APPLIED
	QV	6	5		AGMA 390 QUALITY NUMBER

INVOLUTE GEAR MATHEMATICS
GEOMETRY FACTOR - ACCURATE SPUR GEARS - AS2938-1987.

STP	NAME	PINION	WHEEL	UNIT	DESCRIPTION
6A	RS	160.000	1000000.000	MM	REFERENCE PITCH RADII
6A	SMALLC	5.001	2.632	MM	CUTTER TIP RADII CONSTANT
6A	RU	136.283	883019.583	MM	UNDERCUT RADII
6A	RR	142.819	999976.000	MM	ROOT RADII
6A		NO	NO		UNDERCUT (YES=RU>RR)
6A	BIGRF	151.560	999978.634	MM	RADII TO TOP OF TROCHOID
6	RO	188.000	1000018.000	MM	ORIGINAL TIP RADII
6A	ROM	205.843	1000020.183	MM	MAX ALLOW TIP RADII
6A	CI	.474	.047	MM	INVOLUTE CLEAR. COEFFICIENT
6A		NO	NO		TIP INTERFER. (YES=CI< .04)
6A	CU	.359	.200	MM	BOTTOM CLEAR. COEFFICIENTS
6A		NO	NO		ROOT INTERFER.(YES=CU< .20)
6A	TST	37.107	30.916	MM	REF TRANSV. ARC TOOTH WIDTH
6A	PSIO	0 00'00"	0 00'00"	DEG	TIP HELIX ANGLES
6A	PHIO	36 53'42"	20 00'10"	DEG	TIP TRANSV. PRESSURE ANGLES
6A	TOT	9.069	17.812	MM	TIP TRANSV. ARC TOOTH WIDTH
6A	BETAH	1 22'55"	0 00'02"	DEG	HALF TIP TRANSVERSE ANGLES
6A	TO	9.069	17.812	MM	NORMAL TOP LAND WIDTHS
6A	CW	.453	.891		TOP LAND WIDTH COEFFICIENTS
6A		NO	NO		LAND TOO SMALL (YES=CW<.40)
6A	RMID	169.780	999998.317	MM	RADII TO MID POINT INVOLUTE
6A	ANC	18.898	19.683	MM	CHORDAL HEIGHTS @ RMID
6A	TNC	30.319	32.141	MM	CHORDAL WIDTHS @ RMID
6A	SPAN	157.429	.000	MM	SPAN DIMENSION
6A	STEETH	3.000	.000		OVER NUMBER OF TEETH
6A	EPSLON	1.063	1.147		SLIDE/ROLL RATIOS
6A	MF	.000	.000		FACE CONTACT RATIO
6A	MP	1.480	1.480		TRANSVERSE CONTACT RATIO
6A	MT	1.480	1.480		TOTAL CONTACT RATIO
6A		NO	NO		NON-CONJUGATE ACTN(MT<=1.3)

GEAR GEOMETRY PROGRAM

GEAR SET 1 FULL ITERATION
Demonstration data set for rack and pinion gear set.
GEOMETRY FACTOR - ACCURATE SPUR GEARS - AS2938-1987.

STP	NAME	PINION	WHEEL	UNIT	DESCRIPTION
1	PHIS	20 00'00"	20 00'00"	DEG	TRANSVERSE PRESSURE ANGLE
2	PHIT	20 00'05"	20 00'05"	DEG	OPERATING TRANSV. PRESS ANG
3	C	1000168.000	1000168.000	MM	OPERATING CENTRE DISTANCE
4	RB	150.351	939692.617	MM	BASE RADII
5	R	160.001	1000007.999	MM	OPERATING PITCH RADII
6	R0	188.000	1000018.000	MM	TIP RADII
7	MF	.000	.000		FACE CONTACT RATIO
8	ZB	29.238	29.238	MM	LENGTH OF APPROACH PATH
8	ZA	58.139	58.139	MM	LENGTH OF RECESS PATH
9	Z	87.377	87.377	MM	LENGTH OF LINE OF ACTION
10	ZC	.904	.904	MM	DIST STRESS TO PITCH POINTS
11	CX	.983	.983		CONTACT HEIGHT FACTOR
13	PSIB	0 00'00"	0 00'00"	DEG	BASE HELIX ANGLE
14	CPSI	1.000	1.000		HELICAL OVERLAP FACTOR
15	CC	.161	.161		CURVATURE FACTOR
16	LMIN	235.000	235.000	MM	MIN LGH OF LINES OF CONTACT
17	MNN	1.000	1.000		LOAD SHARING RATIO
18	I	.158	.158		GEOMETRY FACTOR - PITTING
19	PSI	0 00'00"	0 00'00"	DEG	OPERAT PITCH DIA HELIX ANG
20	PHIN	20 00'05"	20 00'05"	DEG	OPERATING NORMAL PRESS ANG
21	NE	16.000	100000.000		VIRTUAL NO. OF SPUR TEETH
21	ROE	188.000	1000018.000	MM	VIRTUAL SPUR TIP RADII
22	BETA	.644	.349	RAD	BASE & TIP RADII INCLD ANGS
22	PHILN	.620	.349	RAD	LOAD ANGLES AT TIP
22	PHIL	.431	.349	RAD	LOAD ANGLES AT HPSTC
23	LAMBAI	.349	.349	RAD	LAMBDA - INITIAL VALUE
24	K1	.479	1.000	MM	CONSTANT - K1
24	K2	1.506	1.350	MM	CONSTANT - K2
24	K3	16.699	50000.327		CONSTANT - K3
25	K4	.916	1.141		VARIABLE - K4
25	K5	.864	1.141	RAD	VARIABLE - K5
25	K6	.848	1.141	RAD	VARIABLE - K6
25	K7	.848	1.141	RAD	VARIABLE - K7
25	K8	.256	.000	RAD	VARIABLE - K8
25	K9	1.104	2.597	MM	VARIABLE - K9
26	TE	41.642	46.714	MM	INSCRIBED PARABOLA - WIDTHS
26	HE	20.669	25.450	MM	INSCRIBED PARABOLA - HEIGHT
27	LAMBDA	.467	.430	RAD	LAMBDA - FINAL VALUES
28	CH	1.000	1.000		HELICAL FACTOR
29	KPSI	1.000	1.000		HELIX ANGLE FACTOR
30	Y	.855	.804		TOOTH FORM FACTORS
31	B	17.183	31.999	MM	OPERATING DEDENDA
31	RF	8.141	4.001	MM	MINIMUM ROOT FILLET RADII
31	H	.180	.180		DOLAN - BROGHAMER FACTOR -H
31	L	.150	.150		DOLAN - BROGHAMER FACTOR -L
31	M	.450	.450		DOLAN - BROGHAMER FACTOR -M
31	KF	1.931	2.080		STRESS CONCENTRATION FACTOR
32	J	.443	.387		GEOMETRY FACTORS - STRENGTH

GEAR GEOMETRY PROGRAM

GEAR SET 1 FULL ITERATION
Demonstration data set for rack and pinion gear set.
THE FOLLOWING TABULATION SHOULD BE INCLUDED ON ENGINEERING DRAWINGS.

ACCURATE SPUR GEARS TO AS2938-1987. MODIFIED ADDENDA

HOB OR COUNTERPART RACK DETAILS	UNIT	PINION	WHEEL
NORMAL MODULE	MM	20.000	20.000
NORMAL PRESSURE ANGLE	DMS	20 00'00"	20 00'00"
TIP RADII	MM	7.600	4.000
STANDARD ADDENDA	MM	25.000	23.313
STANDARD DEDENDA	MM	20.000	18.000

DESIGN DETAILS	UNIT	PINION	WHEEL
NUMBER OF TEETH		16	11
HELIX ANGLE AT REFERENCE PCD	DMS	0 00'00"	0 00'00"
ADDENDUM MODIFICATION COEFFICIENTS		.400	.000
RADIAL TOOTH TRUNCATIONS (TOPPING)	MM	.000	.000
OPERATING NORMAL PRESSURE ANGLE	DMS	20 00'00"	20 00'00"
REFERENCE PITCH CIRCLE DIAMETERS	MM	320.000	63.999
OPERATING PITCH CIRCLE DIAMETERS	MM	320.003	79.997
OPERATING CENTRE DISTANCE	MM	200.000	200.000
OPERATING ADDENDA	MM	27.999	10.001
OPERATING DEDENDA	MM	17.183	31.999
ACTUAL DEPTH OF TEETH	MM	45.181	42.000
NOMINAL DEPTH OF TEETH	MM	45.000	41.313
INVOLUTE CLEARANCES	MM	9.481	.936
BOTTOM CLEARANCES	MM	7.182	4.000
SLIDE/ROLL RATIOS		1.063	1.147
TOTAL CONTACT RATIO		1.480	1.480
GEOMETRY FACTOR - PITTING		.158	.158
GEOMETRY FACTORS - STRENGTH		.443	.387

MACHINING DETAILS	UNIT	PINION	WHEEL
QUALITY NUMBERS (AGMA 390)		6	5
NORMAL THINNING OF TEETH(BACKLASH)	MM	.132	.500
OUTSIDE DIAMETERS	MM	376.000	99.999
ROOT DIA. INCLUDING BACKLASH	MM	285.637	16.000
RADII TO MID POINT OF INVOLUTE	MM	169.780	30.317
CHORDIAL HEIGHTS @ RMID	MM	18.898	19.683
CHORDIAL WIDTHS @ RMID	MM	30.319	32.141
SPAN GAUGE DIMENSIONS	MM	157.429	.000
NUMBER OF TEETH SPANNED		3	0
TOP LAND WIDTHS	MM	9.069	17.812

NOTE: CHORDAL & SPAN DIMENSIONS HAVE BACKLASH INCORPORATED

DESIGNER: ______________________ DATE : __________

8. RECOMMENDED DESIGN GUIDELINES.

Acceptable data entry with out warning messages.

1. TITLE — Gear set description
   68 CHARACTERS.

   Rack & pinion
   1 character either "Y" or "N"

2. NP,NW — Number of pinion, wheel teeth
   For a Pinion & Wheel gear set
   a.) Warning if NP < 10
   b.) Error if NW < NP

   For a Rack & Pinion gear set
   a.) Warning if NP < 10
   b.) Error if RTEETH < 1

3. MN — Normal metric module
   Warning if MN < 1.0
   Warning if MN > 50.0

   Use the standard ISO 53 cutter
   1 character either "Y" or "N"

4. PHIC — Normal pressure angle of cutter
   Warning if PHIC < 14 Deg 30 Min
   Warning if PHIC > 25 Deg

5. HA — Addenda of special cutters
   Warning if HA < 1.00 * MN
   Warning if HA > 1.25 * MN

6. HB — Dedenda of special cutters
   Warning if HB < 0.8 * MN
   Warning if HB > 1.0 * MN

7. RT — Tip radius of special cutters
   Warning if RT < 0.1 * MN
   Error if RT >
   (0.25*PI*MN*COS(PHIC)-HA*SIN(PHIC))/(1.0-SIN(PHIC))

8. PSIS — Helix angle at reference PCD
   Warning if PSIS < 0 Deg.
   Warning if PSIS > 30 Deg.
   Error if PSIS > 45 Deg.

9. F — Face width of narrowest gear
   Spur gears
   Warning if F < 5 * MN
   Warning if F > 10 * MN

   Helical gears
   Warning if F < PI * MN / SIN(PSIS)
   Warning if F > 20 * MN

GEAR GEOMETRY PROGRAM

10. X — Addendum modification coefficient
Warning if X < -0.5
Warning if X > 0.5

11. DLTARO — Truncation applied
Warning if DLTARO < 0.0
Warning if DLTARO > 0.5 * MN

12. BN — Backlash applied
Warning if BN < 0.0
Warning if BN > 0.1 * MN

13. ICODE — Code iteration
One INTEGER either a 0 or 1

14. ROUGH — Accurate spur gears
1 character either "Y" or "N"

15. BUTT(1) — Buttressing of pinion
1 character either "Y" or "N"

BUTT(2) — Buttressing of wheel
1 character either "Y" or "N"

Alter the default nominal values
1 character either "Y" or "N"

16. CIREF — Minimum involute clearance coefficient
Warning if CIREF < 0.04
Warning if CIREF > 0.10

17. CUREF — Minimum root clearance coefficient
Warning if CUREF < 0.20
Warning if CUREF > 0.30

18. CWREF — Minimum top land coefficient
Warning if CWREF < 0.25
Warning if CWREF > 0.50

19. CTREF — Minimum total contact ratio
Spur gears
Warning if CTREF < 1.2
Warning if CTREF > 1.7

Helical gears
Warning if CTREF < 1.4
Warning if CTREF > 4.0

20. QV — Quality numbers
Warning if QV < 6
Warning if QV > 10

These warning limits are safe guards built into the data file creation and edit routine, to ensure the user has entered the intended value. The value is displayed, together with the traditional range for the particular variable being defined. If the same value is confirmed, it is accepted.

GEAR GEOMETRY PROGRAM

9. SAMPLE DATA ENTRY SHEET.

INPUT DATA Required for GEARGEOM.EXE Program.

1. TITLE	______________________			
	Rack & pinion	Yes/No : ______	Cntrs : ______	
2. NP,NW	Number of teeth	Pinion : ______	Wheel : ______	
3. MN	Normal metric module of cutter	(MM) : ______		
	Use the standard ISO 53 cutter	Yes/No : ______	(if Yes go to 8)	

Note:- ISO 53 cutter defined as RT=0.38*MN, HA=1.25*MN, HB=MN, PHIC=20.0 Deg

4. PHIC	Normal pressure angle of special cutter	: ____Deg ____Min ____Sec		
5. HA	Addendum of special cutter (MM)	Pinion : ______	Wheel : ______	
6. HB	Dedenum of special cutter (MM)	Pinion : ______	Wheel : ______	
7. RT	Tip radius of special cutter (MM)	Pinion : ______	Wheel : ______	
8. PSIS	Helix angle at reference PCD	: ____Deg ____Min ____Sec		
9. F	Face width of narrowest gear (MM)	: ______		
10. X	Addendum modification coefficient	Pinion : ______	Wheel : ______	
11. DLTARO	Truncation applied (MM)	Pinion : ______	Wheel : ______	
12. BN	Normal backlash applied (MM)	Pinion : ______	Wheel : ______	
13. ICODE	Code iteration (0=Yes,1=No)	Yes/No : ______		
14. ROUGH	Accurate spur gears (0=Yes,1=No)	Yes/No : ______		
15. BUTT	Buttressing (1=Yes,0=No)	Pinion : ______	Wheel : ______	
	Change default nominal values	Yes/No : ______	(if No go to 20)	
16. CIREF	Minimum involute clearance coefficient	: ______	(Nominal 0.05)	
17. CUREF	Minimum root clearance coefficient	: ______	(Nominal 0.25)	
18. CWREF	Minimum top land width coefficient	: ______	(Nominal 0.40)	
19. CTREF	Minimum for total contact ratio	: ______	(Spur 1.30) (Helical 1.50)	
20. QV	Quality numbers	Pinion : ______	Wheel : ______	

APPENDIX D

LIST OF THE AUTHOR'S PUBLICATIONS

D.1 Gear Design

1. Davey, RJ, "Optimisation of Gears with Particular Reference to Design Parameters", Unpublished M Eng Sc Thesis, Wollongong University College, University of New South Wales, July 1972.

2. Davey, RJ & Wheway, RT, "The Optimum Design of Spur and Helical Gears to AS B61", University Of Wollongong, Department of Mechanical Engineering Design Manual, July 1980.

3. Buchhorn, PA, Davey, RJ, Lopez, MA and Wheway, RT, "The Optimum Design of Spur and Helical Gears to AS B61", Proceedings of the International Symposium on Gearing and Power Transmissions, Tokyo, August-September 1981, Vol II, pp 213-218.

4. Davey, RJ, Lopez, MA and Wheway, RT, "The Influence of Geometric Strength and Zone Factors on the Power Rating of a Pair of Spur Gears", Proceedings IE Aust Symposium on Stress Analysis for Mechanical Design, Sydney, August 1981, pp 99-105.

5. Davey, RJ, Lopez, MA and Wheway, RT, "The Use of CAD and Computer Graphics in the Determination of Geometry Strength Factors for Helical Gears", Proceedings of IE Aust Conference on Computers and Engineering 1983, Sydney, August-September 1983, pp 79-83.

6. Baker, E, Cardillo, P, Davey, R, Hutton, E, Lopez, M, Russell, M, Soady, J, Wheway, RT and Wotherspoon, J, "Geometry Factors I and J in Accordance with AGMA 218.01 for an ISO 53 Generating Rack", University of Wollongong, Department of Mechanical Engineering Report prepared for SAA Committee ME/11, Wollongong, August 1983.

7. Davey, RJ and Wheway, RT, "An Analytical Method for the Determination of the Fundamental Stress Parabola Inscribed within a Gear Tooth", Proceedings of JSPE International Symposium on Design and Synthesis, Tokyo, July 1984, pp 290-295.

8. Cardillo, PC, Davey, RJ and Wheway, RT, "The Computer-Aided Optimum Design of Spur and Helical Gears", Proceedings of JSPE International Symposium on Design and Synthesis, Tokyo, July 1984, pp 328-333.

9. Cardillo, PC, Davey, RJ, Hutton, EA, Lopez, M, Russell, M and Wheway, RT, "Design Data Prepared for the New Australian Gear Standard by a Consortium of Illawarra Engineers", Proceedings of IE Aust Illawarra Group 18th Engineering Conference "Engineering Design Data for the Illawarra", August 1984, 22 pages.

10. Cardillo, PC, Davey, RJ, Lopez, MA, Russell, MJ and Wheway, RT, "A Graphical Method for Determining the Helical Wear Factor I of AGMA 218.01", ASME Paper No 84-DET-167, presented at ASME Fourth International Power Transmission and Gearing Conference, Cambridge, Massachusetts, October 1984.

11. Davey, RJ, Hutton, EA and Wheway, RT, "Mathematical Conundrums of AGMA 218.01", ASME Paper No 84-DET-168, presented at ASME Fourth International Power Transmission and Gearing Conference, Cambridge, Massachusetts, October 1984.

12. Davey, RJ and Wheway, RT, "A Comparison of Theoretical and Experimental Gear Power Ratings", ASME Paper No 84-DET-171, presented at ASME Fourth International Power Transmission and Gearing Conference, Cambridge, Massachusetts, October 1984.

13. Cardillo, PC, Davey, RJ, Lopez, MA and Wheway, RT, "Charts for the Dimensions of the Lewis Equal Strength Parabola", ASME Paper No 84-DET-182, presented at ASME Fourth International Power Transmission and Gearing Conference, Cambridge, Massachusetts, October 1984.

14. Davey, RJ and Wheway, RT, "An Exact Mathematical Solution for the Dimensions of the Lewis Equal Strength Parabola", ASME Paper No 84-DET-183, presented at ASME Fourth International Power Transmission and Gearing Conference, Cambridge, Massachusetts, October 1984.

15. Cardillo, PC, Davey, RJ, Hutton, EA, Lopez, MA, Russell, MJ and Wheway, RT, "Geometry Factor Data for Spur and Helical Gears Designed to the New Australian Gear Standard", Proceedings of the Second World Congress on Gearing, Paris, March 1986, pp 235-246.

16. Davey, RJ and Wheway, RT, "Spur and Helical Gear Design Theory", Paper A of the One Day Workshop "The Design and Rating of Spur and Helical Gears to AS 2938", Wollongong, August 1988.

17. Davey, RJ and Hutton EA, "ASSP-001-1987 - Gear Geometry Software", Standards Association of Australia, Sydney, June 1987.

D.2 Mechanical Engineering Design

1. Davey, RJ, Lopez, MA and Wheway, RT, "An Industrial Application of Computer-Aided Design", Proceedings IE Aust Conference on Computers in Engineering 1981, Melbourne, September 1981, pp 89-93.

2. Davey, RJ and Wotherspoon, JR, "Uprating Skip - Charged Blast Furnaces for Increased Production", Iron and Steel Engineer, Vol 59, No 8, August 1982, pp 37-41.

3. Davey, RJ, Lopez, MA and Wheway, RT, "The Application of Postgraduate Research into Computer-Aided Machine Element Design within the Steel Industry of Port Kembla and the University of Wollongong", Paper 7 of Australian Postgraduate Research Conference, Sydney, February 1983.

4. Davey, RJ, Lopez, MA and Wheway, RT, "An Industrial Case Study of Geometric Modelling as Applied to Skip-Charged Blast Furnaces", Proceedings of JSPE International Symposium on Design and Synthesis, Tokyo, July 1984, pp 65-70.

D.3 Engineering Education

1. Davey, RJ and Wheway, RT, "The Use of a Creative Design Project in the First Year of an Engineering Course", Paper presented at IE Aust Annual Conference, Canberra, April 1972.

2. Davey, RJ, Lopez, MA and Wheway, RT, "University/Industry Co-Operation in The Teaching and Application of Computer-Aided Machine Element Design", Proceedings of AEESEA Regional Conference on Recent Developments in Engineering Education, Bangkok, December 1983, pp 13-28.

3. Davey, RJ and Wheway, RT, "The Use of Creative Design in First Year Engineering Courses", Proceedings of AEESEA Regional Conference on Recent Developments in Engineering Education, Bangkok, December 1983, pp 282-296.

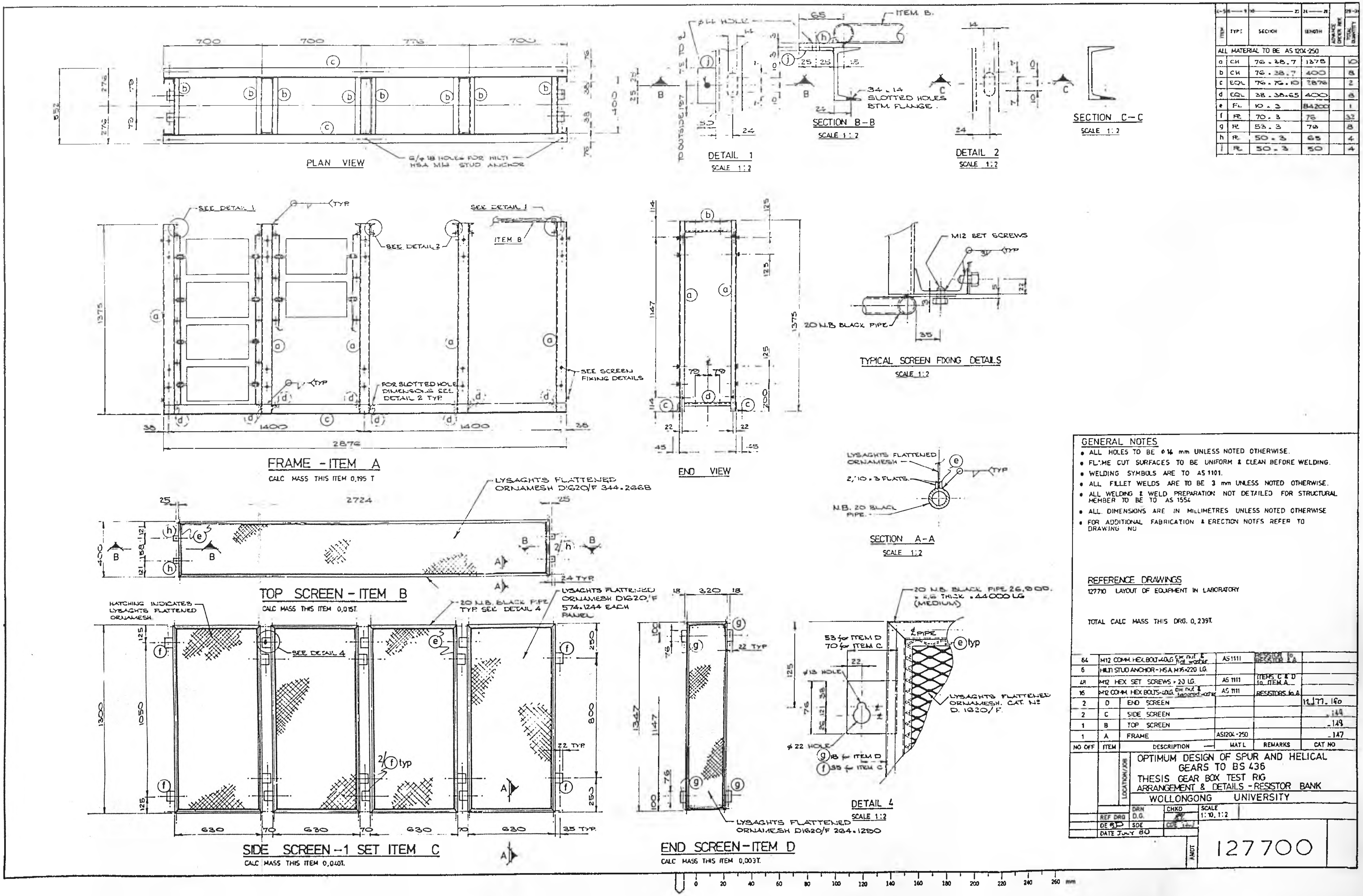
4. Davey, RJ, Lopez, MA and Wheway, RT, "University/Industry Co-Operation in the Teaching and Application of Computer-Aided Machine Element Design", Journal of Engineering Education in Southeast Asia, Vol 14, No 1, June 1984, pp 23-34.

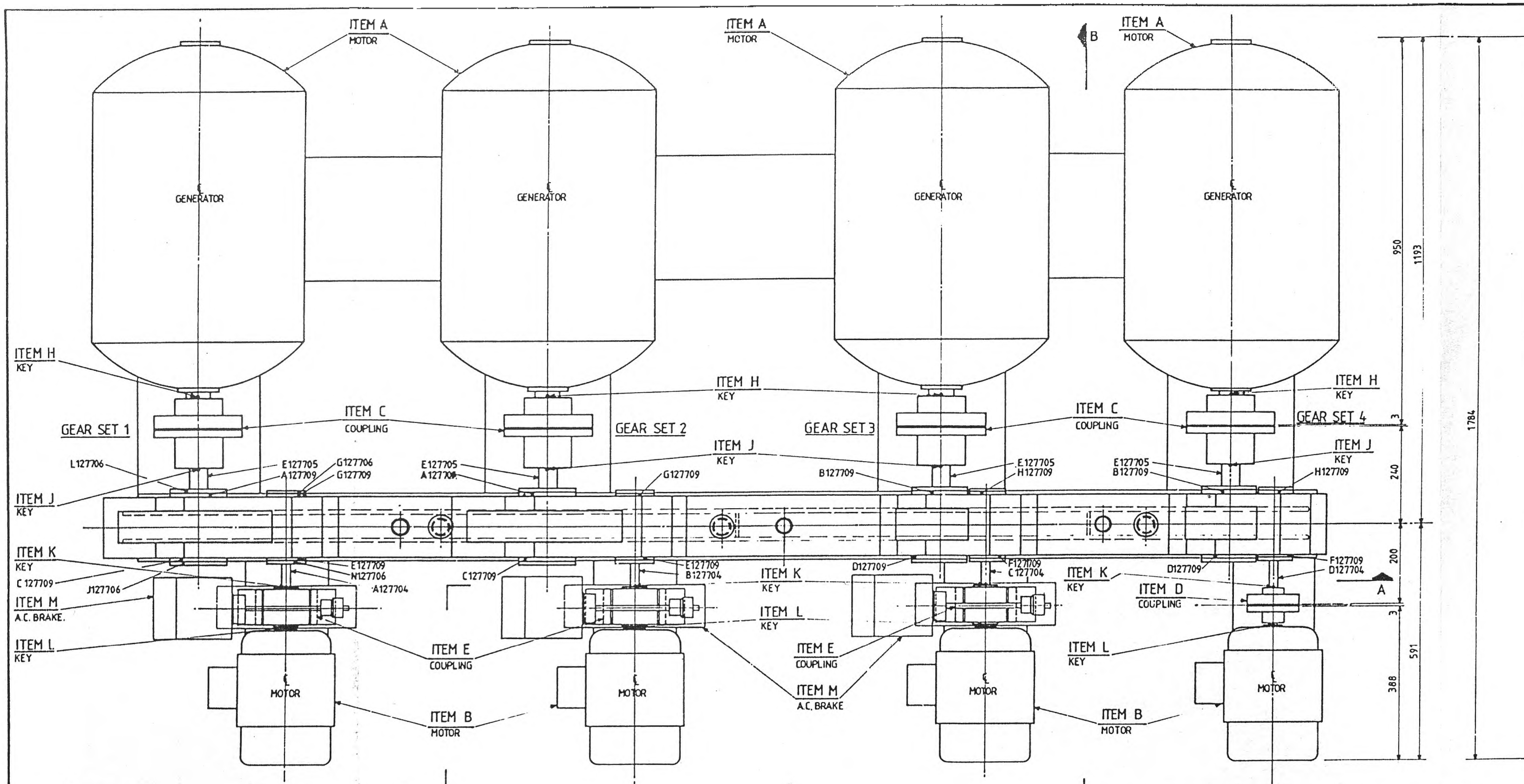
5. Cardillo, PC, Davey, RJ, Lopez, MA and Wheway, RT, "Industry Involvement in the Teaching of Computer-Aided Machine Element Design", Proceedings of the Conference "Teaching Engineering Designers for the 21st Century", Sydney, February 1986, pp 32-37.

6. Davey, RJ and Wheway, RT, "Creative Design Competitions as a Means of Teaching Design in First Year", Proceedings of the Conference "Teaching Engineering Designers for the 21st Century", Sydney, February 1986, pp 46-53.

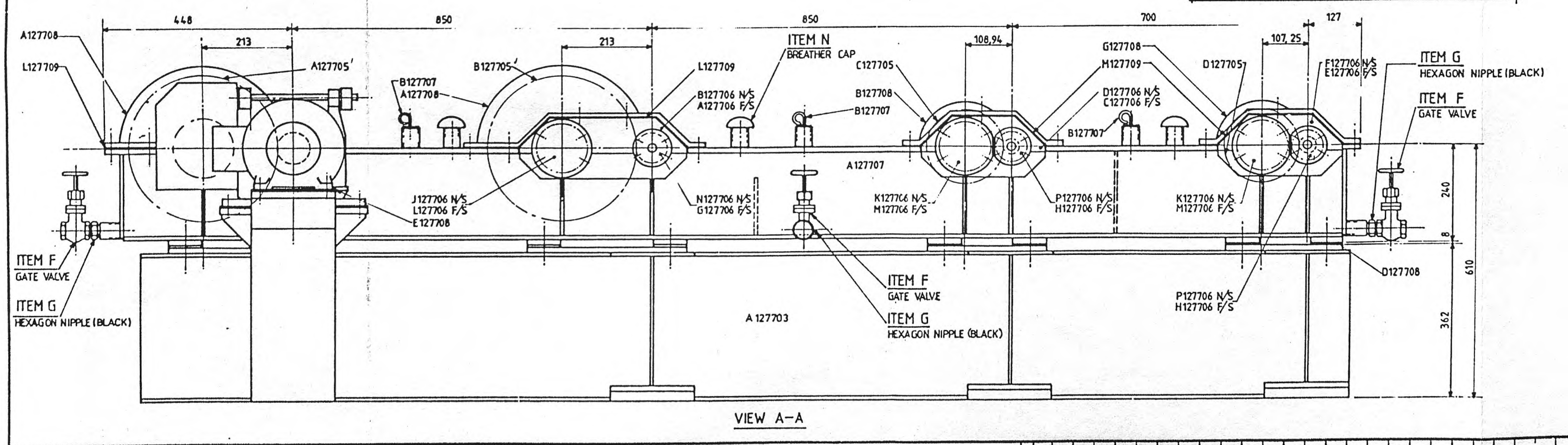
APPENDIX E

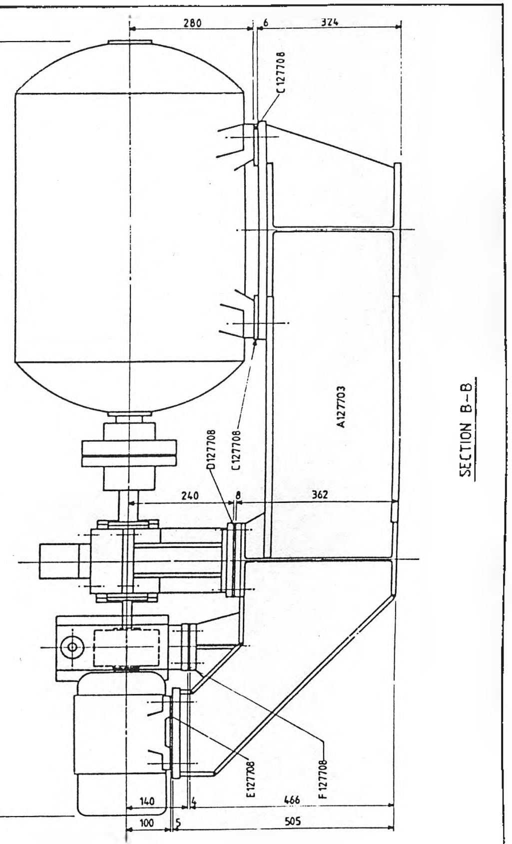
ENGINEERING DRAWINGS OF TEST RIG


PLAN VIEW
6/⌀ 18 HOLES FOR HILTI HSA M16 STUD ANCHOR
DETAIL 1
SCALE 1:2
SECTION B-B
SCALE 1:2
34 × 14 SLOTTED HOLES BTM. FLANGE.
ITEM B.
DETAIL 2
SCALE 1:2
SECTION C-C
SCALE 1:2
ITEM
TYPE
SECTION
LENGTH
ADVANCE ORDER REF.
TOTAL QUANTITY
ALL MATERIAL TO BE AS 1204-250
a CH 76 × 38.7 1375 10
b CH 76 × 38.7 400 8
c EQL 76 × 76 × 10 2876 2
d EQL 38 × 38 × 6.5 400 8
e FL 10 × 3 84200 1
f FL 70 × 3 76 32
g FL 53 × 3 76 8
h FL 50 × 3 65 4
j FL 50 × 3 50 4
SEE DETAIL 1
SEE DETAIL 2
ITEM B
FOR SLOTTED HOLE DIMENSIONS SEE DETAIL 2 TYP.
SEE SCREEN FIXING DETAILS
FRAME - ITEM A
CALC MASS THIS ITEM 0,195 T
END VIEW
M12 SET SCREWS
20 N.B BLACK PIPE
TYPICAL SCREEN FIXING DETAILS
SCALE 1:2
LYSAGHTS FLATTENED ORNAMESH
2/10 × 3 FLATS
N.B. 20 BLACK PIPE
SECTION A-A
SCALE 1:2
GENERAL NOTES
• ALL HOLES TO BE ⌀ 14 mm UNLESS NOTED OTHERWISE.
• FLAME CUT SURFACES TO BE UNIFORM & CLEAN BEFORE WELDING.
• WELDING SYMBOLS ARE TO AS 1101.
• ALL FILLET WELDS ARE TO BE 3 mm UNLESS NOTED OTHERWISE.
• ALL WELDING & WELD PREPARATION NOT DETAILED FOR STRUCTURAL MEMBER TO BE TO AS 1554
• ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE
• FOR ADDITIONAL FABRICATION & ERECTION NOTES REFER TO DRAWING NO
LYSAGHTS FLATTENED ORNAMESH D1620/F 344.2668
TOP SCREEN - ITEM B
CALC MASS THIS ITEM 0,015T.
HATCHING INDICATES LYSAGHTS FLATTENED ORNAMESH
20 N.B. BLACK PIPE TYP. SEE DETAIL 4
LYSAGHTS FLATTENED ORNAMESH D1620/F 574.1244 EACH PANEL
SEE DETAIL 4
SIDE SCREEN - 1 SET ITEM C
CALC MASS THIS ITEM 0,040T.
LYSAGHTS FLATTENED ORNAMESH D1620/F 284.1250
END SCREEN - ITEM D
CALC MASS THIS ITEM 0,003T.
20 N.B. BLACK PIPE 26.9 OD. × 2.6 THICK × 44000 LG (MEDIUM)
53 for ITEM D
70 for ITEM C
⌀ 13 HOLE
⌀ 22 HOLE
18 for ITEM D
35 for ITEM C
LYSAGHTS FLATTENED ORNAMESH. CAT. No D. 1620/F.
DETAIL 4
SCALE 1:2
REFERENCE DRAWINGS
127710 LAYOUT OF EQUIPMENT IN LABORATORY
TOTAL CALC MASS THIS DRG. 0,239T.
64 M12 COMM. HEX. BOLT × 40 LG c/w nut & flat washer AS 1111 RESISTOR to RESISTOR A
6 HILTI STUD ANCHOR - HSA M16 × 220 LG.
48 M12 HEX SET SCREWS × 20 LG. AS 1111 ITEMS C & D to ITEM A
16 M12 COMM. HEX BOLTS × 40 LG. c/w nut & tapered washer AS 1111 RESISTORS to A
2 D END SCREEN 127771.150
2 C SIDE SCREEN .149
1 B TOP SCREEN .149
1 A FRAME AS1204-250 .147
NO OFF ITEM DESCRIPTION MAT'L REMARKS CAT NO
LOCATION/JOB
OPTIMUM DESIGN OF SPUR AND HELICAL GEARS TO BS 436
THESIS GEAR BOX TEST RIG
ARRANGEMENT & DETAILS - RESISTOR BANK
WOLLONGONG UNIVERSITY
REF DRG
DRN D.G.
CHKD
SCALE 1:10, 1:2
DE
SDE
CDE
DATE JULY 80
AMDT
127700



THIS DRAWING TO BE READ IN CONJUNCTION
WITH DRAWING NO 127702
FOR COUPLING BORE DETAILS REF 127705



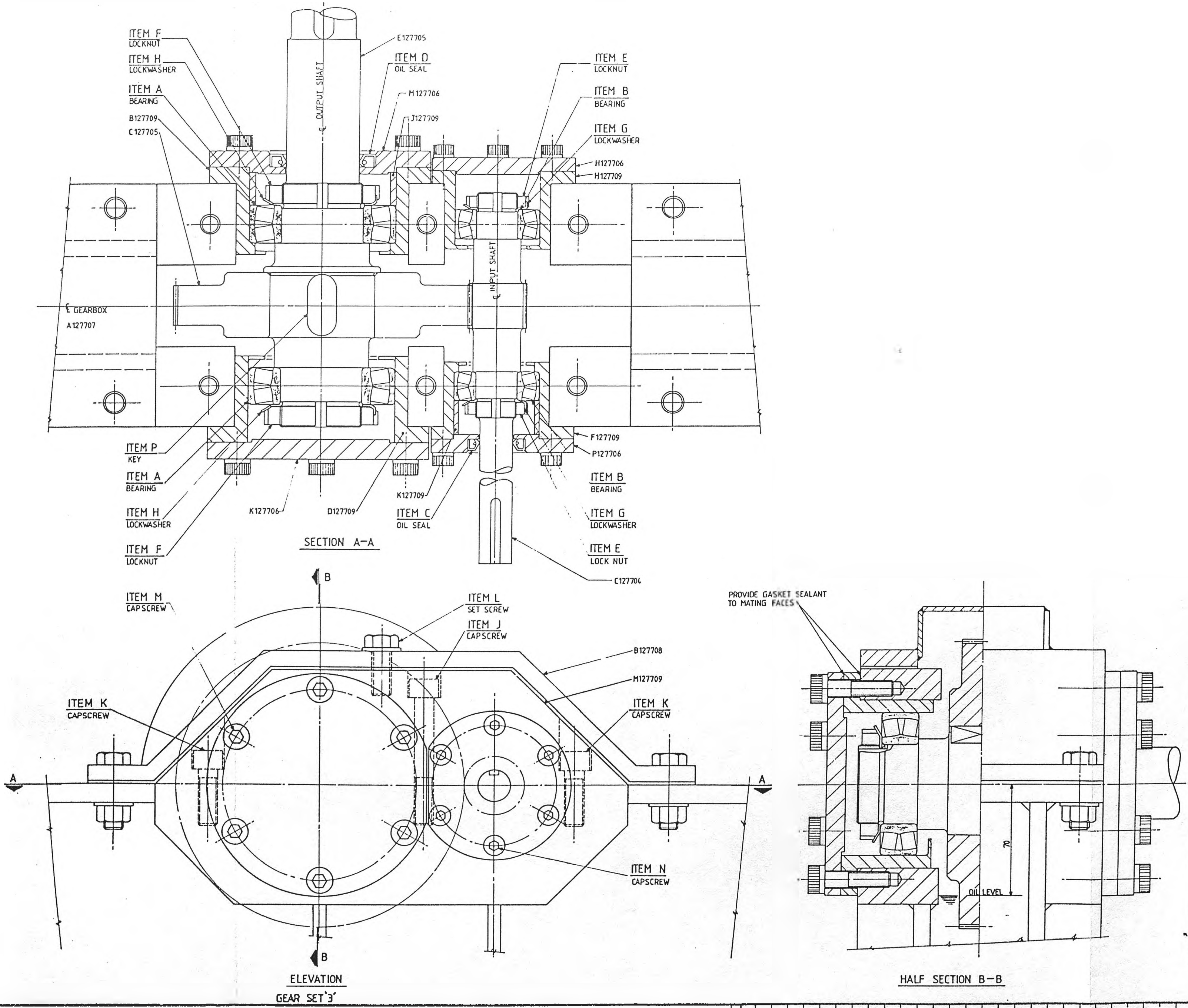


16	COMM. HEX. BOLT M20 x 90 LG. C/W nuts & flat washer	AS 1111	ITEM A TO A127703	
16	COMM. HEX. BOLT M12 x 60 LG. C/W nuts & flat washer	AS 1111	A127707 TO A127703	
20	COMM. HEX. BOLT M12 x 45 LG. C/W nuts & flat washer	AS 1111	B127708 TO A127707	
28	COMM. HEX. BOLT M10 x 60 LG. C/W nuts & flat washer	AS 1111	ITEM B & H TO A127703	

NO OFF	ITEM	DESCRIPTION —	MATL	REMARKS	CAT NO
3	N	BREATHER CAP			12.128_189
3	M	A.C. BRAKE.		PERIGRIP SIZE J	
4	L	8 x 7 x 40 LG. RECT. PAR'L. KEY			.188
4	K	6,35 x 6,35 x 40 LG. SQ. KEY			.187
4	J	12 x 8 x 65 LG. RECT. PAR'L KEY			.186
4	H	5/8" x 7/16" x 3" LG. RECT. PAR'L. KEY			.185
3	G	25NB HEXAGON NIPPLE (BLACK)	BS 1740		.184
3	F	25NB GATE VALVE		SCREWED	.183
3	E	COUPLING - FLEXI-DRIVE BRAKE DRUM		MORSE FRB6	.182
1	D	COUPLING - CONE RING		D.BROWN. MC-030	.181
4	C	COUPLING - CONE RING		D.BROWN. MC-070	.180
4	B	2,2 KW A.C. MOTOR.		SQUIRREL CAGE	
4	A	5 HP D.C. MOTOR		CONVERT TO GENERATOR	

LOCATION/JOB	OPTIMUM DESIGN OF SPUR AND HELICAL GEARS TO BS 436 THESIS GEAR BOX - TEST RIG ARRANGEMENT — SH.1.
	WOLLONGONG UNIVERSITY
REF DRG	DRN RF
DE	SDE
	CHKD
	CDE
DATE JUNE 80'	SCALE 1:5
AMDT	127701

0 20 40 60 80 100 120 140 160 180 200 220 240 260 mm



REFERENCE DRAWINGS

127701 - ARRANGEMENT SH.1
127703 - GEARBOX SUPPORT FRAME
127704 - FIRST MOTION SHAFTS
127705 - MECHANICAL DETAILS
127706 - GEARBOX DETAILS - SH. 1.
127707 - GEARBOX DETAILS - SH. 2.
127708 - GEARBOX DETAILS - SH. 3
127709 - GEARBOX DETAILS - SH. 4.

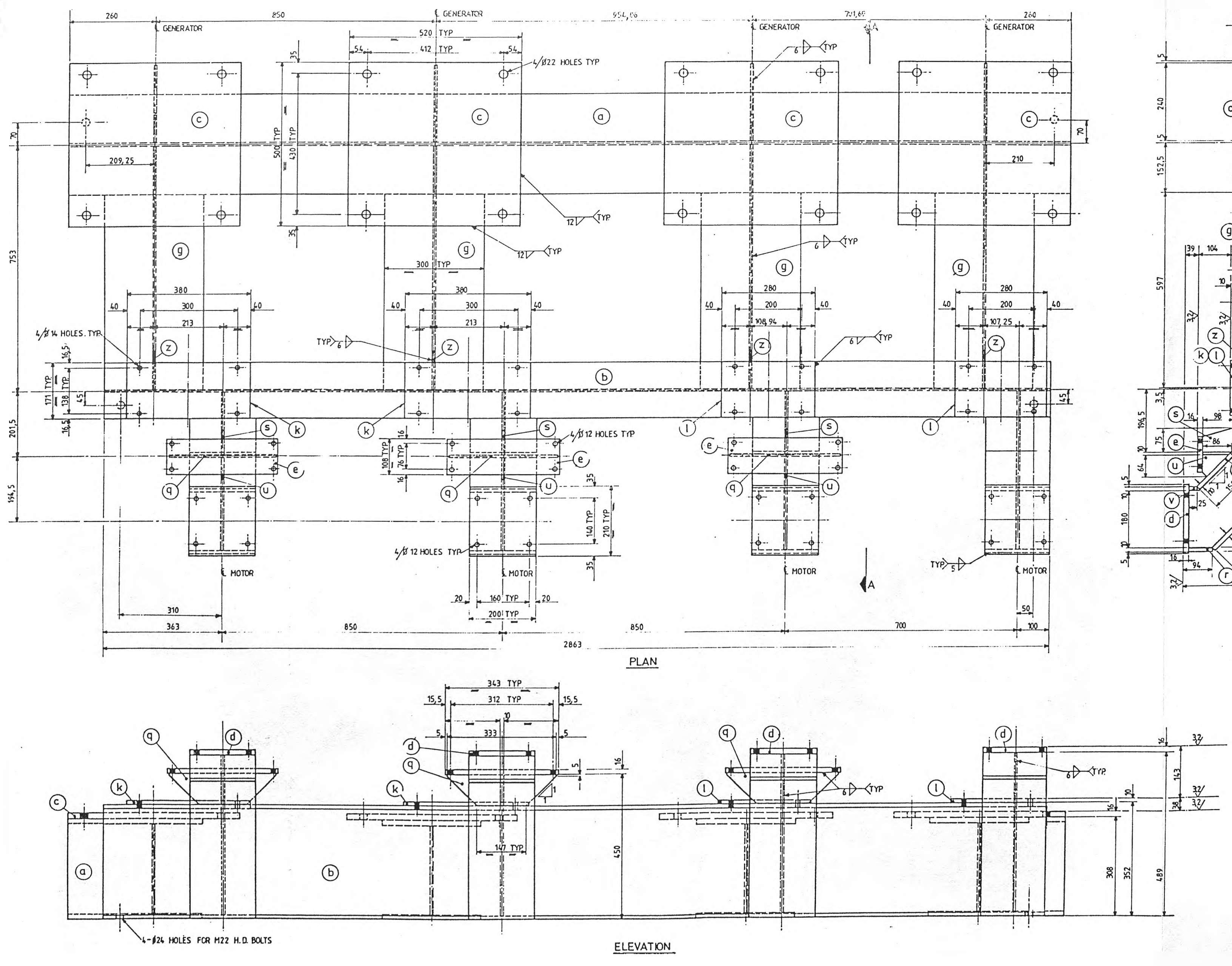
THIS DRAWING TO BE READ IN CONJUNCTION WITH DRAWING NO. 127701

NO OFF	ITEM	DESCRIPTION	MATL	REMARKS	CAT NO
4	P	18 x 11 x 20 LG. RECT PAR'L KEY			
48	N	SOCKET HEAD CAPSCREW M8 x 35			
48	M	SOCKET HEAD CAPSCREW M10 x 40			
12	L	HEXAGON HD. SET SCREW M12 x 30		C/W WASHER	
16	K	SOCKET HEAD CAPSCREW M12 x 35			
8	J	SOCKET HEAD CAPSCREW M12 x 80			
8	H	'SKF' MB10 LOCKWASHER			
8	G	'SKF' MB 5 LOCKWASHER			
8	F	'SKF' KM 10 LOCKNUT			
8	E	'SKF' KM 5 LOCKNUT			
4	D	'GACO' OIL SEAL Ø45 x Ø60 x 8			
4	C	'GACO' OIL SEAL Ø20 x Ø35 x 7			
8	B	'SKF' 22205C BEARING			
8	A	'SKF' 22210C BEARING			

LOCATION/JOB
OPTIMUM DESIGN OF SPUR AND HELICAL GEARS TO BS 436
THESIS GEAR BOX – TEST RIG ARRANGEMENT. SH 2.
WOLLONGONG UNIVERSITY

REF DRG
DRN R.F.
CHKD
SCALE 1:1
DET
SDE RW
CDE
DATE JUNE 80
AMDT
127702

0 20 40 60 80 100 120 140 160 180 200 220 240 260 mm

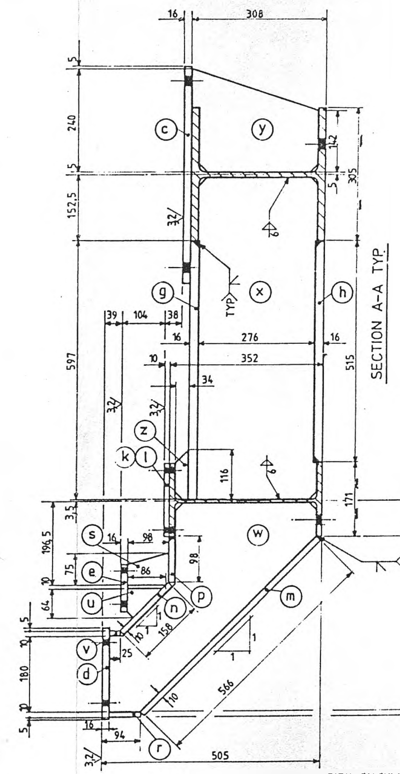


PLAN

ELEVATION

SUPPORT FRAME — ITEM A

MASS=0,935 TONNE



SECTION A–A TYP.

ITEM	TYPE	SECTION	LENGTH	ADVANCE ORD REF	TOTAL QUANTITY
ALL MATERIAL TO BE AS 1204-250					
a	UC	310 UC 97	3026		1
b	UB	360 UB 45	2863		1
c	PL	500 x 20	520		4
d	PL	200 x 20	210		4
e	PL	108 x 20	343		3
f					
g	PL	300 x 16	597		4
h	PL	300 x 16	515		4
j					
k	PL	171 x 12	380		2
l	PL	171 x 12	280		2
m	PL	200 x 10	566		4
n	PL	158 x 10	200		4
p	PL	98 x 10	200		4
q	PL	86 x 10	333		3
r	PL	78 x 10	200		4
s	PL	75 x 10	98		3
t					
u	PL	64 x 10	76		3
v	PL	25 x 10	200		4
w	PL	479 x 8	482		4
x	PL	278 x 8	744		4
y	PL	240 x 8	292		4
z	PL	34 x 8	116		4

TOTAL CALCULATED MASS THIS DRAWING=0,935 TONNE

GENERAL NOTES

- ALL HOLES TO BE ø 12 mm UNLESS NOTED OTHERWISE.
- FLAME CUT SURFACES TO BE UNIFORM & CLEAN BEFORE WELDING.
- WELDING SYMBOLS ARE TO AS 1101.
- ALL FILLET WELDS ARE TO BE 12 mm UNLESS NOTED OTHERWISE.
- ALL WELDING & WELD PREPARATION NOT DETAILED FOR STRUCTURAL MEMBER TO BE TO AS 1554
- ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.
- FOR ADDITIONAL FABRICATION & ERECTION NOTES REFER TO DRAWING NO

REFERENCE DRAWINGS

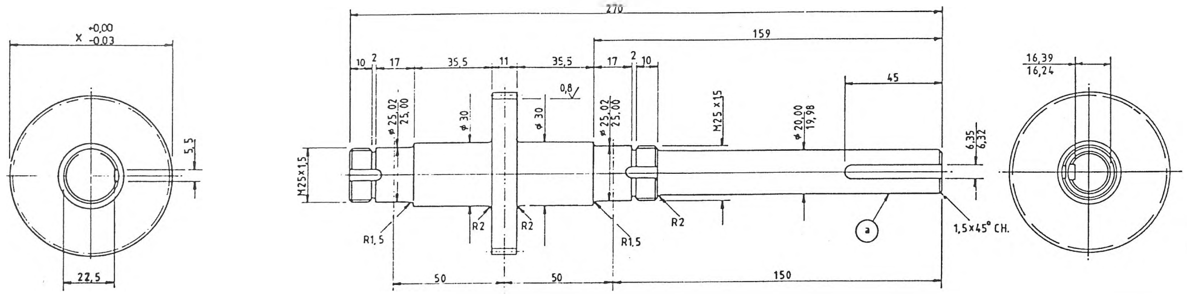
127701 ARRANGEMENT SH. 1.

NO OFF	ITEM	DESCRIPTION	MAT L	REMARKS	CAT NO
1	A	SUPPORT FRAME	AS 1204-250		11.J15.1/0

OPTIMUM DESIGN OF SPUR AND HELICAL GEARS TO BS 436
THESIS GEAR BOX - TEST RIG
GEAR BOX SUPPORT FRAME DETAILS
WOLLONGONG UNIVERSITY

DRN R.F. | CHKD | SCALE 1:5
REF DRG | DE | SDE | CDE
DATE MAY 80

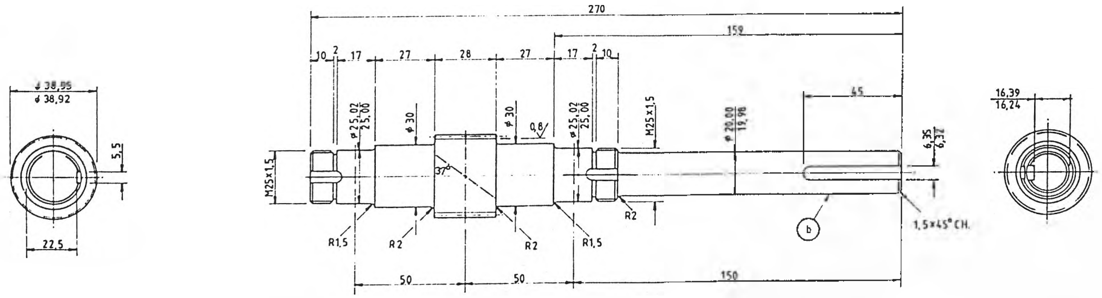
127703



ITEM	X
A	73,64
B	73,0

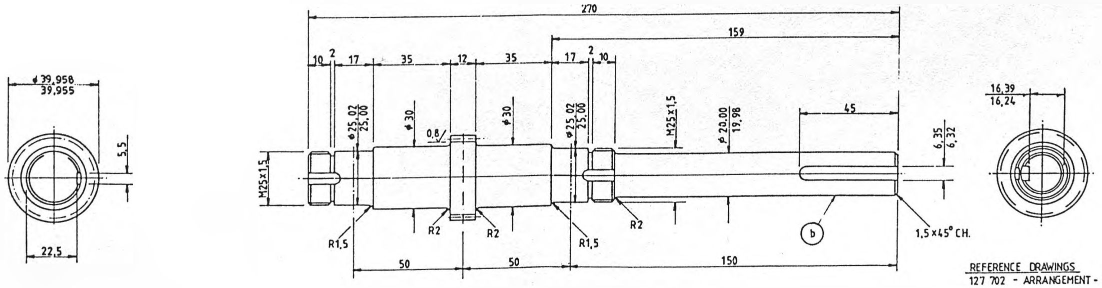
FIRST MOTION SHAFT FOR GEAR SET ONE - ITEM A
FIRST MOTION SHAFT FOR GEAR SET TWO - ITEM B

ALL OVER U.N.O CALC. MASS = 1,3 Kg.



FIRST MOTION SHAFT FOR GEAR SET THREE - ITEM C

1,6 ALL OVER U.N.O. CALC. MASS = 1,06 Kg.



FIRST MOTION SHAFT FOR GEAR SET FOUR - ITEM D

1,6 ALL OVER U.N.O. CALC. MASS = 1,03 Kg

REFERENCE DRAWINGS
127 702 - ARRANGEMENT - SHEET 2

TOTAL CALC. MASS THIS DRG. = 0,047 TONNES

FOR GENERAL NOTES, REFER DRG. 127 707

METRIC GEAR DATA - MODIFIED ADDENDUM - AS B61 - HELICAL - RIGHT HAND HELIX

DESCRIPTION				ITEM C	
NUMBER OF TEETH				29	
REVS. PER MINUTE				1440	
HELIX ANGLE (SIGMA)				37,0°	
NORMAL DIAMETRAL MODULE				1,00	
TRANSVERSE DIAMETRAL MODULE				1,252	
NORMAL CIRCULAR PITCH				3,142	
TRANSVERSE CIRCULAR PITCH				3,934	
AXIAL PITCH				5,220	
EFFECTIVE FACEWIDTH OF GEAR				25,04	
OVERLAP RATIO				2,398	
NORMAL PRESS ANGLE OF CUTTER (DEGREES)				20,0°	
TRANSVERSE PRESS ANGLE OF CUTTER (DEGREES)				24°30'2"	
THEORETICAL P.C.D.				36,312	
CORRECTION FACTORS KP KW (CL.43)				0,32	
WHOLE DEPTH OF TOOTH				2,25	
ADDENDUM (1)				1,32	
DEDENDUM (1)				0,93	
TIP OR BLANK DIAMETER (2)				38,95	
NOMINAL ROOT DIAMETER				34,52	
THEORETICAL CENTRE DISTANCE				108,936	
RUNNING CENTRE DISTANCE (1)(CL.43B)				108,94	
BOTTOM CLEARANCE (3)				0,25	
PITCH LINE VELOCITY (M/MIN.)				164,3	
CLASS OF GEAR (CL.1)				C	
BACKLASH TOOTH THINNING ON EACH GEAR (CL.46)				0,058	
CUTTER SETTING CORRECTION				0,320	
CONSTANT CHORD HEIGHT (4)				1,03	
CONSTANT CHORD THICKNESS (4)				1,53	
MATERIAL SPECIFICATIONS (APPENDIX A)				EN8Q	
MIN TENSILE STRENGTH (MPa)				616	
MIN. BRINELL N° (SURFACE) (B.S.240)				179	
MIN. BRINELL N° (CORE) (B.S.240)				179	
EQUIVALENT RUNNING TIME (HRS/DAY)(CL.69 & 70)				24	
RATED CAPACITY STRENGTH (K.W.) (5)(CL.64)				2,1	
RATED CAPACITY WEAR (K.W.)(5)(CL.64)				2,0	

METRIC GEAR DATA - MODIFIED ADDENDUM - A.S.B.61 - SPUR GEARING

DESCIPTION	FORMULAE	ITEM A	ITEM B		ITEM D
NUMBER OF TEETH	t, T	71	71		24
REVS. PER MINUTE	n, N	1440	1440		1440
DIAMETRAL MODULE	m	1,0	1,0		1,5
PRESS. ANGLE OF CUTTER (DEGREES)	Υ	20,0°	20,0°		20,0°
THEORETICAL P.C.D.	d = tm, D = Tm	71,0	71,0		36,0
CORRECTION FACTORS	KP, KW (SEE CLAUSE 43)	0,32	0		0,319
WHOLE DEPTH OF TOOTH	H = 2,25m	2,25	2,25		3,375
ADDENDUM (1)	a = (1+KP)m A = (1+KW)m	1,320	1,0		1,979
DEDENDUM (1)	b = H-a B = H-A	0,930	1,25		1,396
TIP OR BLANK DIAMETER (2)	j = d+2a J = D+2A	73,64	73,0		39,958
NOMINAL ROOT DIAMETER	i = d-2b I = D-2B	69,14	68,5		33,208
THEORETICAL CENTRE DISTANCE	C = 0,5m (t+T)	213,0	213,0		107,25
RUNNING CENTRE DISTANCE (1)	(SEE CLAUSE 43b)	213,0	213,0		107,25
BOTTOM CLEARANCE (3)	c = C-(J+i)/2 = C-(I+j)/2	0,25	0,25		0,375
PITCH LINE VELOCITY (m/min)	v = πdn/1000 = πDN/1000	321,2	321,2		162,9
CLASS OF GEAR	(SEE CLAUSE 1)	B	B		C
BACKLASH TOOTH THINNING / GEAR	q = 2σ σw = m(T+60)/10,000+0,0254	0,077	0,077		0,076
CUTTER SETTER CORRECTION	ΔP = KPm ΔW = KWm	0,320	0,0		0,479
CONSTANT CHORD HEIGHT (4)	h = (1-0,125π sin 2Υ)m + Δcos² Υ	1,030	0,75		1,544
CONSTANT CHORD THICKNESS (4)	g = 0,5m(π cos² Υ) + Δ sin 2Υ - q	1,516	1,310		2,312
MATERIAL SPECIFICATIONS	(SEE APPENDIX A)	EN8Q	EN8Q		EN8Q
MINIMUM TENSILE STRENGTH (MPa)		616	616		616
MINIMUM BRINELL N° (SURFACE)	(SEE B.S. 240)	179	179		179
MINIMUM BRINELL N° (CORE)	(SEE B.S. 240)	179	179		179
EQUIV. RUNNING TIME (hrs./day)	(SEE CLAUSES 69 & 70)	24	24		0,5
RATED CAPACITY STRENGTH (K.W.)(5)	(SEE CLAUSE 64)	2,0	2,0		2,3
RATED CAPACITY WEAR (K.W.)(5)	(SEE CLAUSE 64)	2,0	2,0		2,0

(1) Ref. Merritt section 16.15 for non standard centre distance.
(2) If t < 17 ref. note clause 43.
(3) Bottom clearance = 0,25m if centre distance standard.
(4) At point of contact (see clause 44)
(5) Power calculated at speed of revolution of each gear.
(6) All linear dimensions in millimetres.

NO OFF	ITEM	DESCRIPTION	MAT'L	REMARKS	CAT NO
1	D	SHAFT - PINION	EN8Q	SPUR	11.J11.194
1	C	SHAFT - PINION	EN8Q	HELICAL	193
1	B	SHAFT - PINION	EN8Q	SPUR	192
1	A	SHAFT - PINION	EN8Q	SPUR	191

LOCATION/JOB

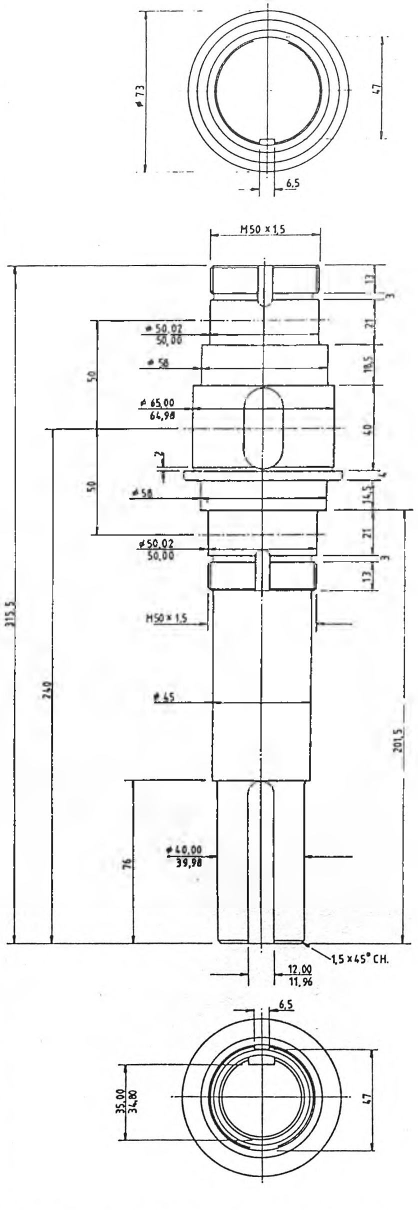
OPTIMUM DESIGN OF SPUR AND HELICAL GEARS TO BS 436
THESIS GEARBOX TEST RIG
FIRST MOTION SHAFTS

WOLLONGONG UNIVERSITY

REF DRG	DRN TLH	CHKD	SCALE 1:1
DE'RD	SDE	CDE	
DATE MAY 80			

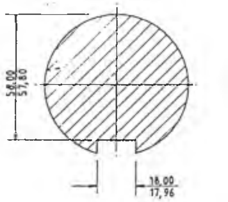
AMDT

127704

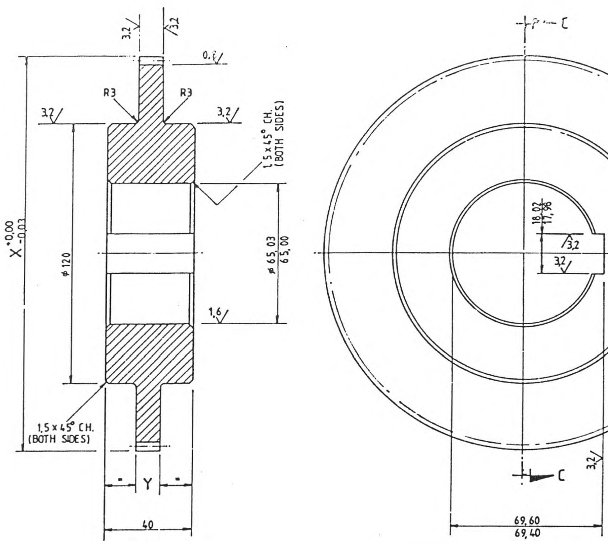


SECOND MOTION SHAFT FOR ALL GEAR SETS
ITEM E

1,6 ALL OVER ALL RADII R1,5 CALC. MASS = 4,8 Kg.



SECTION THROUGH KEYWAY ON ⌀ 65,00 / 64,98



SECTION C C

ITEM	X	Y	DESCRIPTION
A	356,36	10,8	GEAR - SET ONE
B	357,0	10,8	GEAR - SET TWO
C	182,92	25,04	GEAR - SET THREE
D	180.542	11,57	GEAR - SET FOUR

GEAR WHEEL - ITEM A	AS DRAWN & NOTED	CALC. MASS = 10,0 Kg.
GEAR WHEEL - ITEM B	AS DRAWN & NOTED	CALC. MASS = 10,0 Kg.
GEAR WHEEL - ITEM C	AS DRAWN & NOTED	CALC. MASS = 5,4 Kg.
GEAR WHEEL - ITEM D	AS DRAWN & NOTED	CALC. MASS = 3,8 Kg.

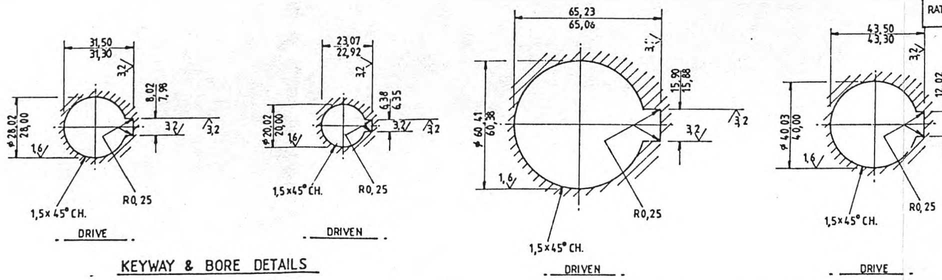
METRIC GEAR DATA - MODIFIED ADDENDUM - AS B61 - HELICAL - LEFT HAND HELIX

DESCRIPTION				ITEM C	
NUMBER OF TEETH				145	
REVS. PER MINUTE				288	
HELIX ANGLE (SIGMA)				37,00°	
NORMAL DIAMETRAL MODULE				1,0	
TRANSVERSE DIAMETRAL MODULE				1,252	
NORMAL CIRCULAR PITCH				3,142	
TRANSVERSE CIRCULAR PITCH				3,934	
AXIAL PITCH				5,220	
EFFECTIVE FACEWIDTH OF GEAR				25,04	
OVERLAP RATIO				2,398	
NORMAL PRESS ANGLE OF CUTTER (DEGREES)				20,0	
TRANSVERSE PRESS. ANGLE OF CUTTER (DEGREES)				24° 30' 2"	
THEORETICAL P.C.D.				181,56	
CORRECTION FACTORS KP KW (CL 43)				-0,32	
WHOLE DEPTH OF TOOTH				2,25	
ADDENDUM (1)				0,68	
DEDENDUM (1)				1,57	
TIP OR BLANK DIAMETER (2)				182,92	
NOMINAL ROOT DIAMETER				178,42	
THEORETICAL CENTRE DISTANCE				108,936	
RUNNING CENTRE DISTANCE (1)(CL 43B)				108,94	
BOTTOM CLEARANCE (3)				0,25	
PITCH LINE VELOCITY (M/MIN)				164,3	
CLASS OF GEAR (CL 1)				C	
BACKLASH TOOTH THINNING ON EACH GEAR (CL 46)				0,082	
CUTTER SETTING CORRECTION				-0,320	
CONSTANT CHORD HEIGHT (4)				0,465	
CONSTANT CHORD THICKNESS (4)				1,100	
MATERIAL SPECIFICATIONS (APPENDIX A)				EN 8	
MIN TENSILE STRENGTH (MPa)				540	
MIN BRINELL Nº (SURFACE) (B.S. 240)				152	
MIN. BRINELL Nº (CORE) (B.S. 240)				152	
EQUIVALENT RUNNING TIME (HRS/DAY) (CL. 69 & 70)				24,6	
RATED CAPACITY STRENGTH (K.W.) (5) (CL. 64)				2,0	
RATED CAPACITY WEAR (K.W.) (5) (CL. 64)				2,0	

METRIC GEAR DATA - MODIFIED ADDENDUM - A.S.B. 61 - SPUR GEARING

DESCIPTION	FORMULAE	ITEM A	ITEM B		ITEM D
NUMBER OF TEETH	t, T	355	355		119
REVS. PER MINUTE	n, N	288	288		290
DIAMETRAL MODULE	m	1,00	1,00		1,5
PRESS. ANGLE OF CUTTER (DEGREES)	Ψ	20,00	20,00		20,0
THEORETICAL P.C.D.	d = tm, D = Tm	355,0	355,0		178,5
CORRECTION FACTORS	KP, KW (SEE CLAUSE 43)	-0,32	0		-0,319
WHOLE DEPTH OF TOOTH	H = 2,25m	2,25	2,25		3,375
ADDENDUM (1)	a = (1+KP)m A = (1+KW)m	0,68	1,0		1,021
DEDENDUM (1)	b = H-a B = H-A	1,57	1,25		2,354
TIP OR BLANK DIAMETER (2)	j = d+2a J = D+2A	356,36	357,0		180,542
NOMINAL ROOT DIAMETER	i = d-2b I = D-2B	351,86	352,5		173,792
THEORETICAL CENTRE DISTANCE	C = 0,5m (t+T)	213,0	213,0		107,25
RUNNING CENTRE DISTANCE (1)	(SEE CLAUSE 43b)	213,0	213,0		107,25
BOTTOM CLEARANCE (3)	c = C-(J+i)/2 = C-(I+j)/2	0,25	0,25		0,375
PITCH LINE VELOCITY (m/min)	v = πdn/1000 = πDN/1000	321,2	321,2		162,9
CLASS OF GEAR	(SEE CLAUSE 1)	B	B		C
BACKLASH TOOTH THINNING / GEAR	q = 2σ σw = m(T+60)/10,000+0,0254	0,134	0,134		0,104
CUTTER SETTER CORRECTION	ΔP = KPm ΔW = KWm	-0,32	0		-0,479
CONSTANT CHORD HEIGHT (4)	h = (1-0,125π sin 2Ψ)m + Δcos²Ψ	0,465	0,75		0,698
CONSTANT CHORD THICKNESS (4)	g = 0,5m(π cos²Ψ) + Δ sin 2Ψ - q	1,048	1,253		1,668
MATERIAL SPECIFICATIONS	(SEE APPENDIX A)	EN 8	EN 8		EN 8
MINIMUM TENSILE STRENGTH (MPa)		540	540		540
MINIMUM BRINELL Nº (SURFACE)	(SEE B.S. 240)	152	152		152
MINIMUM BRINELL Nº (CORE)	(SEE B.S. 240)	152	152		152
EQUIV. RUNNING TIME (hrs./day)	(SEE CLAUSES 69 & 70)	24,0	24,0		0,5
RATED CAPACITY STRENGTH (K.W.) (5)	(SEE CLAUSE 64)	2,0	2,0		2,1
RATED CAPACITY WEAR (K.W.) (5)	(SEE CLAUSE 64)	2,0	2,0		2,0

FOR OTHER DATA REFER DRAWING Nº 127704.



KEYWAY & BORE DETAILS

FOR ITEM E127 701
FOR ITEM D127 701

KEYWAY & BORE DETAILS - FOR ITEM C127 701

REFERENCE DRAWINGS
127 702 - ARRANGEMENT - SHEET 2
TOTAL CALC. MASS THIS DRG. = 0,0484 TONNES
FOR GENERAL NOTES REFER DRG. 127 707

NO OFF	ITEM	DESCRIPTION	MAT'L	REMARKS	CAT NO
4	E	SHAFT	EN8Q		12.J13.199
1	D	GEAR WHEEL	EN8	SPUR	.198
1	C	GEAR WHEEL	EN8	HELICAL	.197
1	B	GEAR WHEEL	EN8	SPUR	.196
1	A	GEAR WHEEL	EN8	SPUR	.195

LOCATION/JOB
OPTIMUM DESIGN OF SPUR AND HELICAL GEARS TO BS 436
THESIS GEARBOX TEST RIG
MECHANICAL DETAILS
WOLLONGONG UNIVERSITY

REF DRG | DRN TLH | CHKD | SCALE 1:1
DE | SDE | CDE
DATE MAY 80

AMDT

127705

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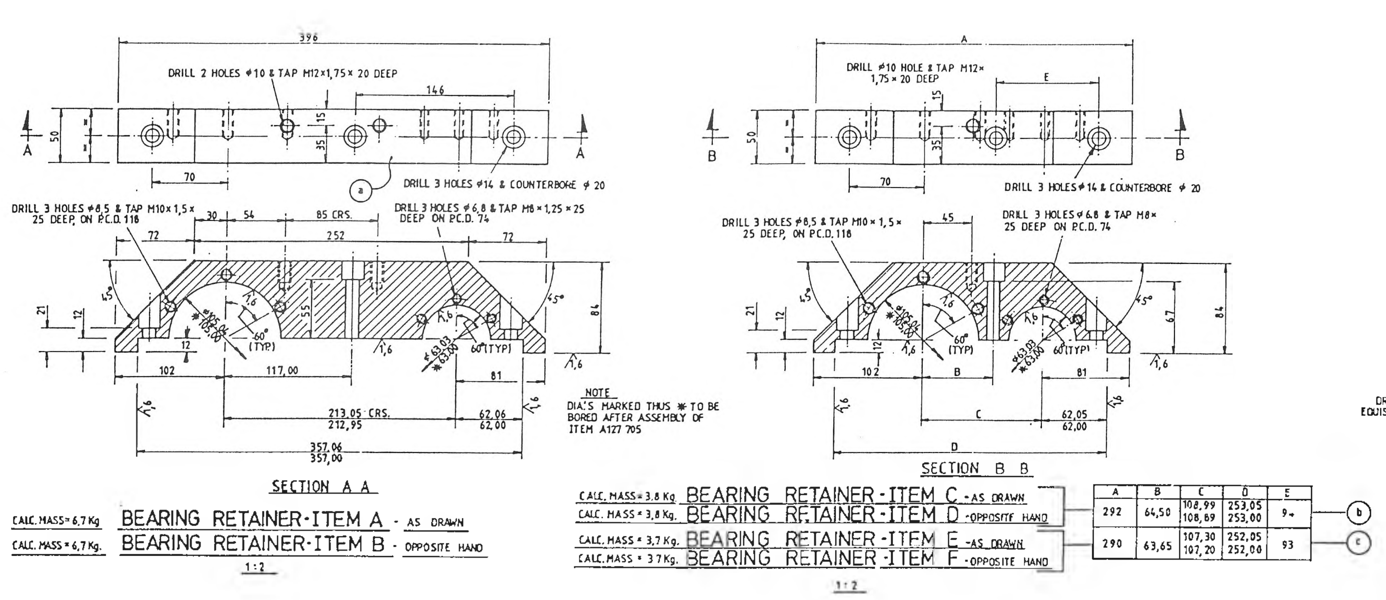
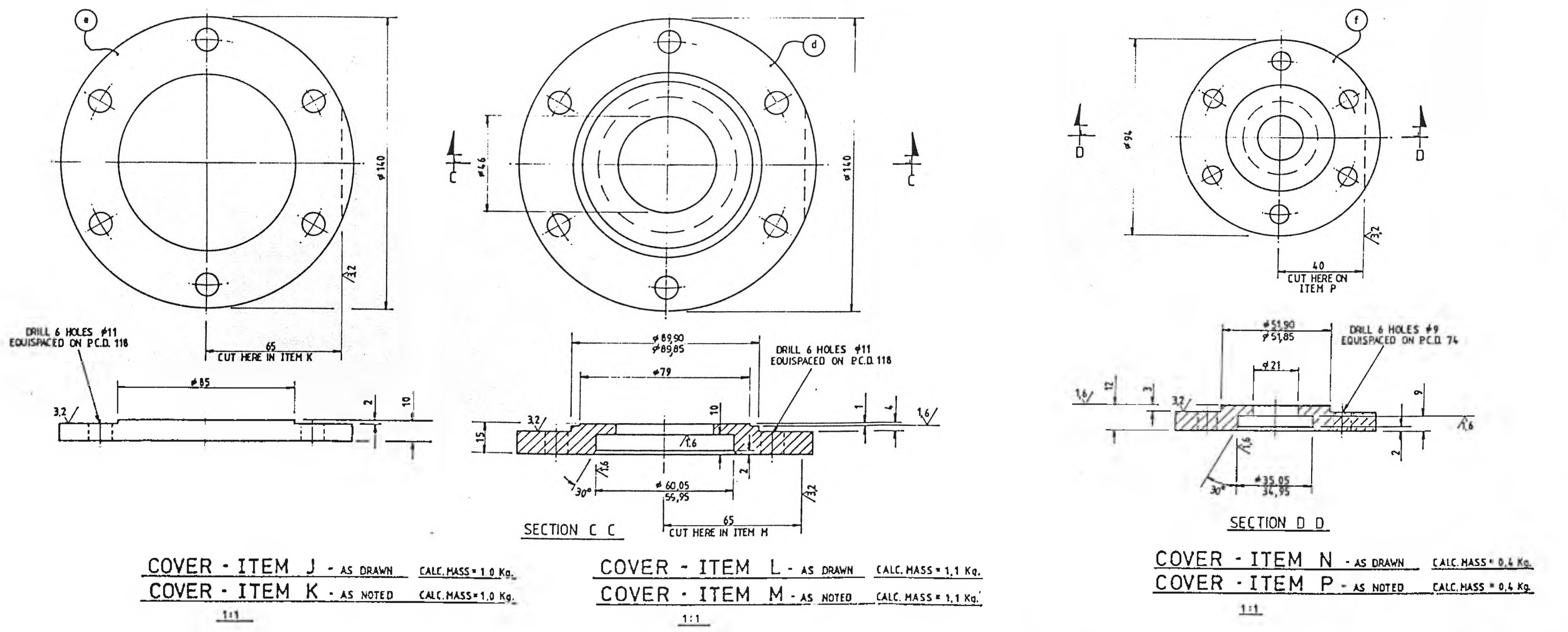
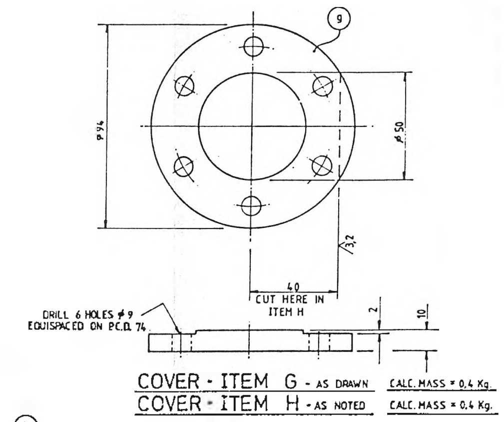






ITEM	TYPE	SECTION	LENGTH	ADVANCE ORDER REF	TOTAL QUANTITY
ALL MATERIAL TO BE AS1204-250					
a	PL	70 × 50	396		4
b	PL	90 × 50	292		2
c	PL	90 × 50	290		2
d	PL	140 × 16	140		4
e	PL	94 × 12	94		4
f	PL	140 × 10	140		4
g	PL	94 × 10	94		4

TOTAL CALC. MASS THIS DRG. = 0,0534 TONNES

GENERAL NOTES

ALL HOLES SHALL BE ⌀ 11mm UNLESS NOTED OTHERWISE.

DIMENSIONS TO MACHINED SURFACES SHALL BE ±0,1 mm UNLESS OTHERWISE NOTED.

FINISH 1,6 Ra ALL OVER UNLESS NOTED OTHERWISE.

STRESS RELIEVE AT 550 — 600°C BEFORE MACHINING.

KEYS & KEYWAYS SHALL BE IN ACCORDANCE WITH BS4235.

ALL SHARP EDGES SHALL BE REMOVED

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS OTHERWISE NOTED

ALL WELDING SHALL BE COMPLETED BEFORE MACHINING.

WELDING SYMBOLS ARE TO AS1101.

ALL WELDING PREPARATION & PROCEDURES NOT SHOWN SHALL CONFORM TO AS 1554

REFERENCE DRAWINGS
127 702 - ARRANGEMENT - SHEET 2

NO OFF	ITEM	DESCRIPTION — MATL		REMARKS	CAT NO
2	P	COVER	AS1204-250		[illegible].113
2	N	COVER	AS1204-250		.112
2	M	COVER	AS1204-250		.111
2	L	COVER	AS1204-250		.110
2	K	COVER	AS1204-250		.109
2	J	COVER	AS1204-250		.108
2	H	COVER	AS1204-250		.107
2	G	COVER	AS1204-250		.106
1	F	BEARING RETAINER	AS1204-250		.105
1	E	BEARING RETAINER	AS1204-250		.104
1	D	BEARING RETAINER	AS1204-250		.103
1	C	BEARING RETAINER	AS1204-250		.102
2	B	BEARING RETAINER	AS1204-250		.101
2	A	BEARING RETAINER	AS1204-250		.100

LOCATION/JOB

OPTIMUM DESIGN OF SPUR AND HELICAL GEARS TO BS436
THESIS. GEARBOX TEST RIG
GEARBOX DETAILS SHT 1 of 4

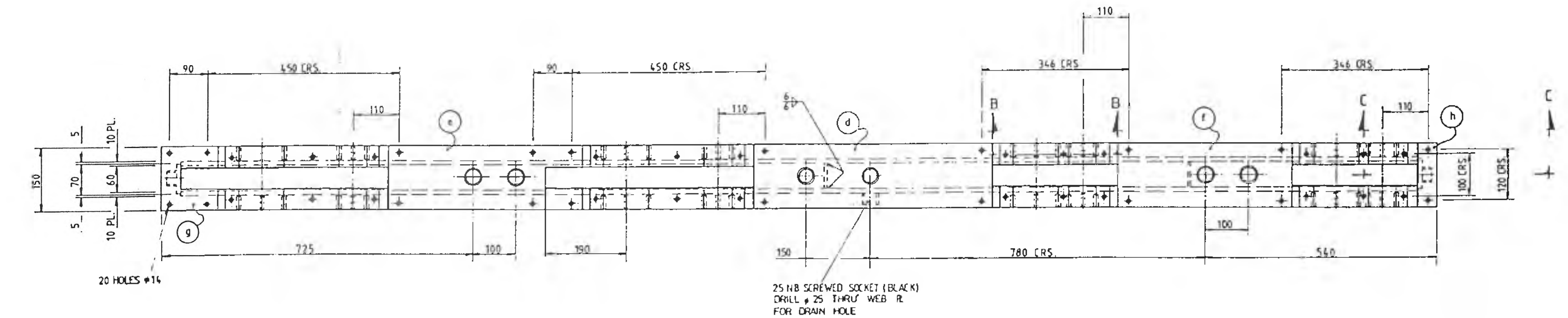
WOLLONGONG UNIVERSITY

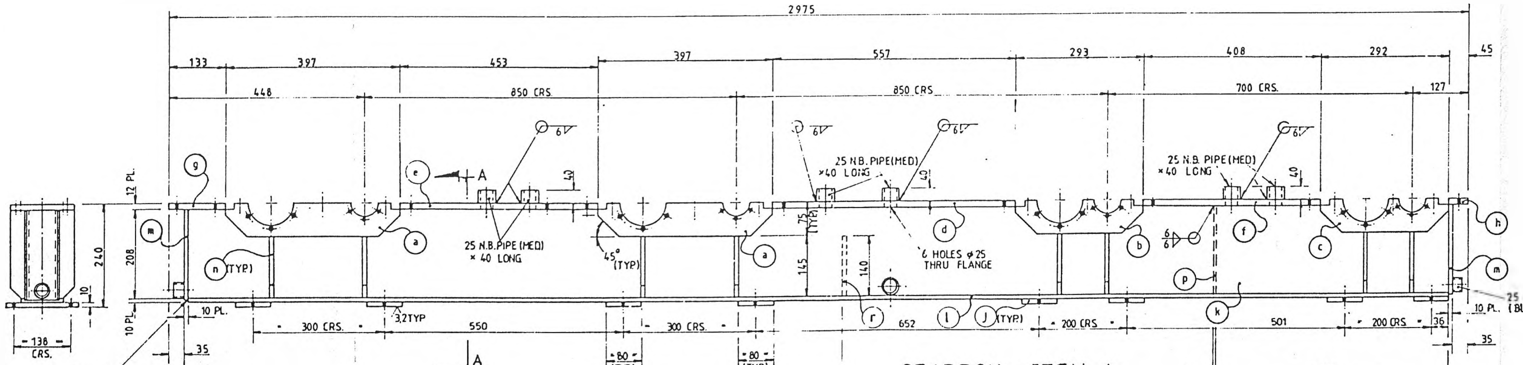
REF DRG | DRN TLH | CHKD | SCALE 1:1 & 1:2
DATE MAY 80

AMDT

127706

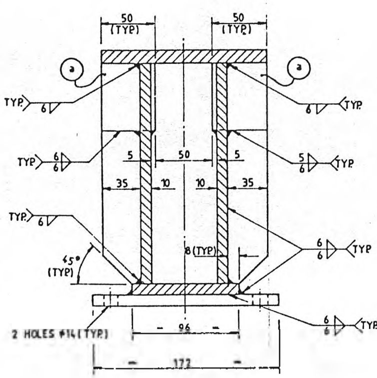
ITEM	TYPE	SECTION	LENGTH	ADVANCE ORDER REF	TOTAL QUANTITY
ALL MATERIAL TO BE AS1204-250					
a	PL	30 × 50	397		4
b	PL	80 × 50	293		2
c	PL	80 × 50	292		2
d	PL	150 × 12	557		1
e	PL	150 × 12	453		1
f	PL	150 × 12	408		1
g	PL	150 × 12	133		1
h	PL	150 × 12	45		1
j	PL	80 × 12	172		6
k	PL	208 × 10	2885		2
l	PL	96 × 10	2885		1
m	PL	70 × 10	213		2
n	PL	35 × 10	145		16
p	PL	60 × 6	208		1
r	PL	60 × 6	140		1
s	PL	30 × 3	30		3
t	RND	25 DIA.	20		3
u	RND	6 DIA.	334		3





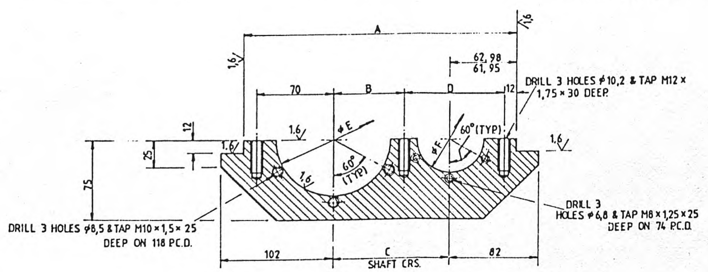
GEARBOX - ITEM A

1:5 CALC. MASS = 207,8 Kg.



SECTION A A

1:2

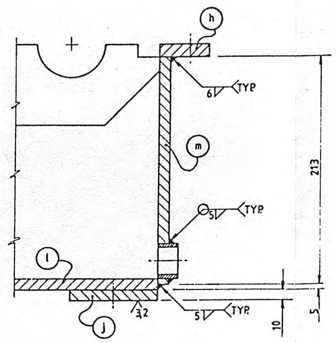


SECTION B B

1:2

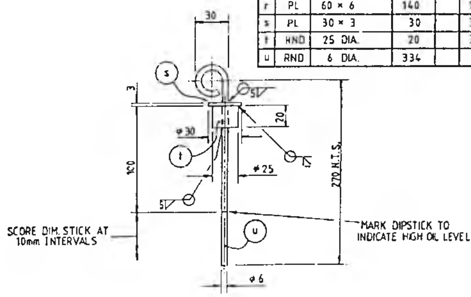
	A	B	C	D	E	F
a	356,98 356,95	117,00	213,05 212,95	146	105,04 105,00	63,03 63,00
b	252,98 252,95	65,00	108,99 108,89	94	105,04 105,00	63,03 63,00
c	251,98 251,95	64,00	107,30 107,20	93	105,04 105,00	63,03 63,00

NOTE DIA'S E & F TO BE BORED AFTER ASSEMBLY OF ITEMS A,B,C,D,E & F 127 706



SECTION C C

1:2



DIPSTICK - ITEM B

1:2 CALC. MASS = 0,5 Kg

TOTAL CALC. MASS THIS DRG. = 0,2088 TONNES

GENERAL NOTES

ALL HOLES SHALL BE ø 14 mm UNLESS NOTED OTHERWISE.

DIMENSIONS TO MACHINED SURFACES SHALL BE ± 0,1 mm UNLESS OTHERWISE NOTED.

FINISH 1,6 Ra ALL OVER UNLESS NOTED OTHERWISE.

STRESS RELIEVE AT 550 — 600°C BEFORE MACHINING.

KEYS & KEYWAYS SHALL BE IN ACCORDANCE WITH BS4235.

ALL SHARP EDGES SHALL BE REMOVED.

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS OTHERWISE NOTED.

ALL WELDING SHALL BE COMPLETED BEFORE MACHINING.

WELDING SYMBOLS ARE TO AS1101.

ALL WELDING PREPARATION & PROCEDURES NOT SHOWN SHALL CONFORM TO AS 1554

REFERENCE DRAWINGS

127 701 - ARRANGEMENT - SHEET 1

NO OFF	ITEM	DESCRIPTION	MAT'L	REMARKS	CAT NO
3	B	DIPSTICK	AS1204-250		11 323 215
1	A	GEARBOX	AS1204-250		214

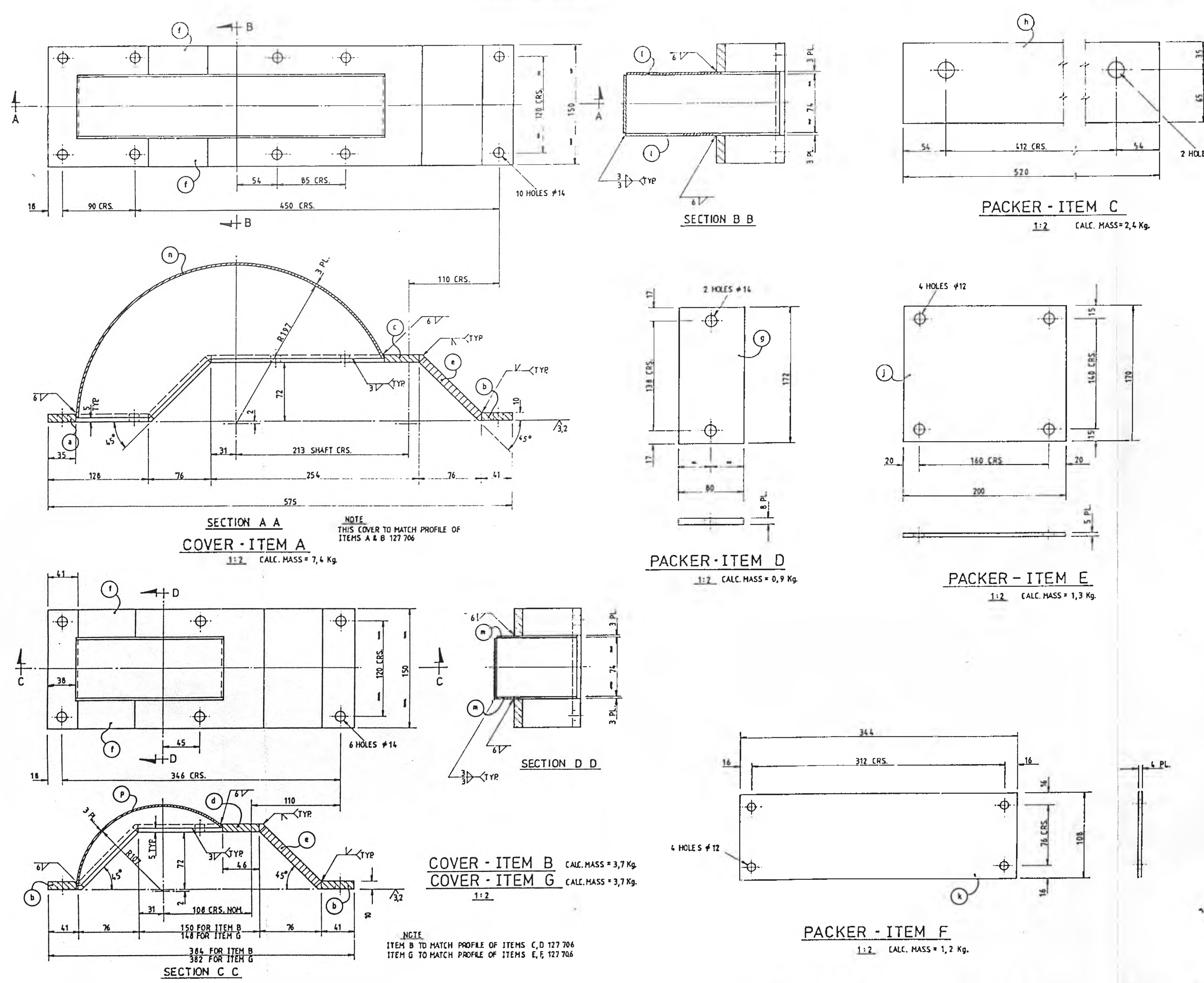
OPTIMUM DESIGN OF SPUR AND HELICAL GEARS TO BS436
THESIS GEARBOX TEST RIG
GEARBOX DETAILS SHT 2 of 4

WOLLONGONG UNIVERSITY

DRN TLH | CHKD | SCALE 1:2 & 1:5
REF DRG | DE | SDE | CDE
DATE MAY 80

AMDT

127707



ITEM	TYPE	SECTION	LENGTH	ADVANCE ORDER REF	TOTAL QUANTITY
ALL MATERIAL TO BE AS1204-250					
a	PL	128 × 12	150		2
b	PL	41 × 12	150		6
c	PL	150 × 10	254		2
d	PL	150 × 10	150		2
e	PL	97 × 10	150		4
f	PL	35 × 10	97		8
g	PL	80 × 8	172		8
h	PL	100 × 6	520		6
j	PL	170 × 5	200		4
k	PL	108 × 4	344		3
l	PL	190 × 3	376		4
m	PL	100 × 3	180		4
n	PL	74 × 3	535		2
p	PL	74 × 3	255		2

GENERAL NOTES

- ALL HOLES TO BE ⌀ 14 mm UNLESS NOTED OTHERWISE.
- FLAME CUT SURFACES TO BE UNIFORM & CLEAN BEFORE WELDING.
- WELDING SYMBOLS ARE TO AS 1101.
- ALL FILLET WELDS ARE TO BE 3 mm UNLESS NOTED OTHERWISE.
- ALL WELDING & WELD PREPARATION NOT DETAILED FOR STRUCTURAL MEMBER TO BE TO AS 1554.
- ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.
- FOR ADDITIONAL FABRICATION & ERECTION NOTES REFER TO DRAWING NO.

TOTAL CALC. MASS THIS DRG = 0,0547 TONNES

REFERENCE DRAWINGS
127 702 - ARRANGEMENT - SHEET 2

NO OFF	ITEM	DESCRIPTION	MAT'L	REMARKS	CAT NO
1	G	COVER	AS1204-250		12 J13.112
3	F	PACKER	AS1204-250		.111
4	E	PACKER	AS1204-250		.110
8	D	PACKER	AS1204-250		.119
8	C	PACKER	AS1204-250		.118
1	B	COVER	AS1204-250		.117
2	A	COVER	AS1204-250		.116

LOCATION/JOB

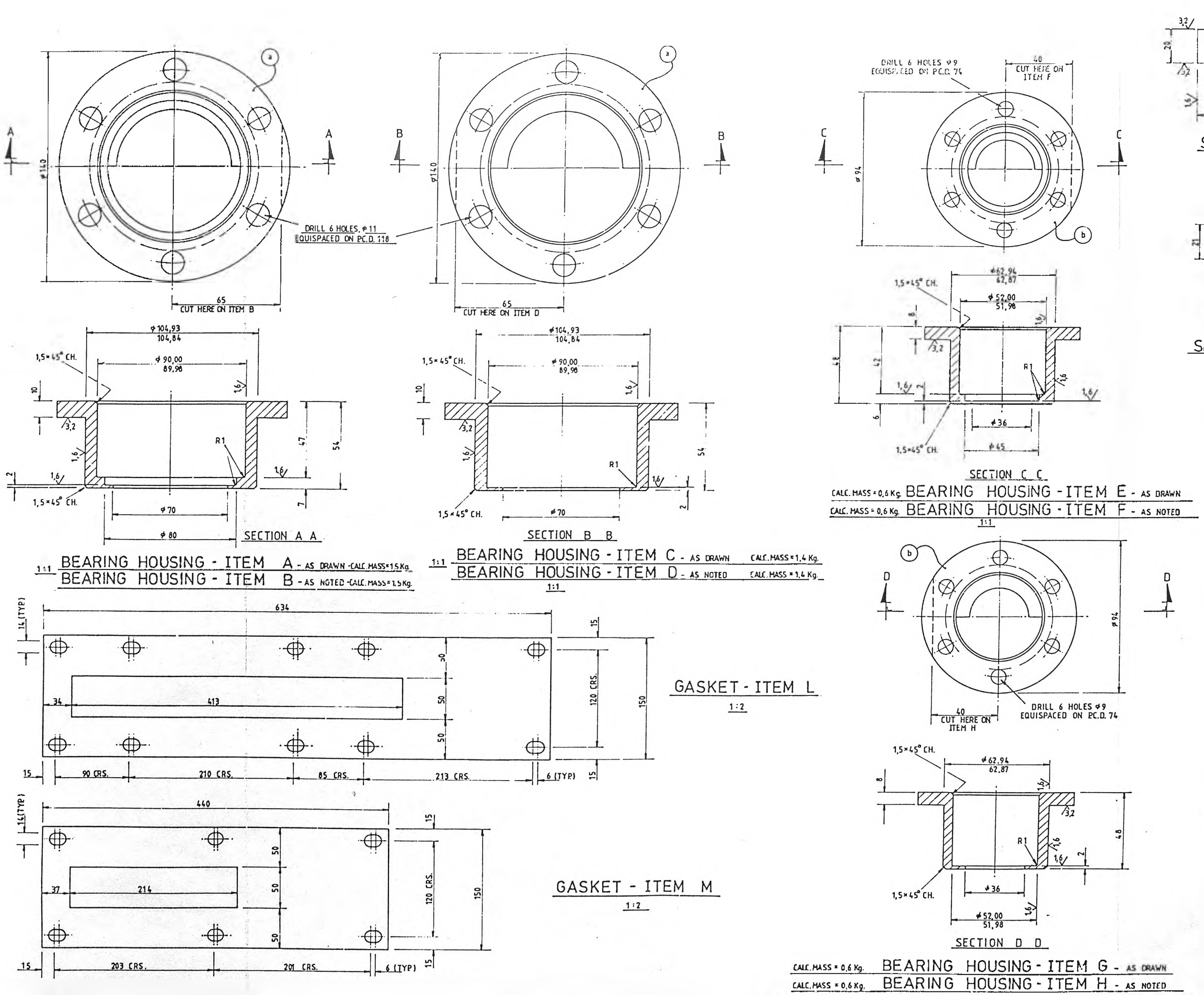
OPTIMUM DESIGN OF SPUR AND HELICAL GEARS TO BS 436
THESIS GEARBOX TEST RIG
GEARBOX DETAILS SHT. 3 of 4

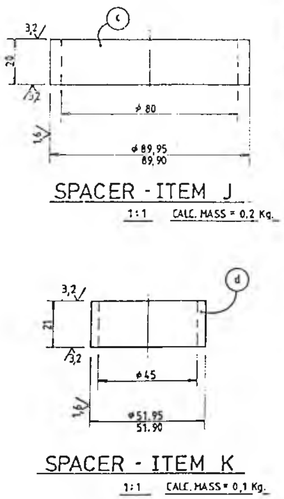
WOLLONGONG UNIVERSITY

REF DRG | DRN TLH | CHKD | SCALE 1:2
DE | SDE | CDE
DATE MAY 80

AMDT | 127 708

ITEM	TYPE	SECTION	LENGTH	ADVANCE ORDER REF	TOTAL QUANTITY
ALL MATERIAL TO BE AS1204-250					
a	RND	140 DIA.	54		8
b	RND	100 DIA.	48		8
c	RND	100 DIA.	20		4
d	RND	56 DIA.	21		4





REFERENCE DRAWINGS
127 702 - ARRANGEMENT - SHEET 2

TOTAL CALC. MASS THIS DRG. = 0, 0164 TONNES

GENERAL NOTES

ALL HOLES SHALL BE ⌀ 11 mm UNLESS NOTED OTHERWISE.

DIMENSIONS TO MACHINED SURFACES SHALL BE ± mm UNLESS OTHERWISE NOTED.

FINISH 3,2 Ra ALL OVER UNLESS NOTED OTHERWISE.

STRESS RELIEVE AT 550 – 600°C BEFORE MACHINING.

KEYS & KEYWAYS SHALL BE IN ACCORDANCE WITH BS4235.

ALL SHARP EDGES SHALL BE REMOVED.

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS OTHERWISE NOTED

ALL WELDING SHALL BE COMPLETED BEFORE MACHINING.

WELDING SYMBOLS ARE TO AS 1101.

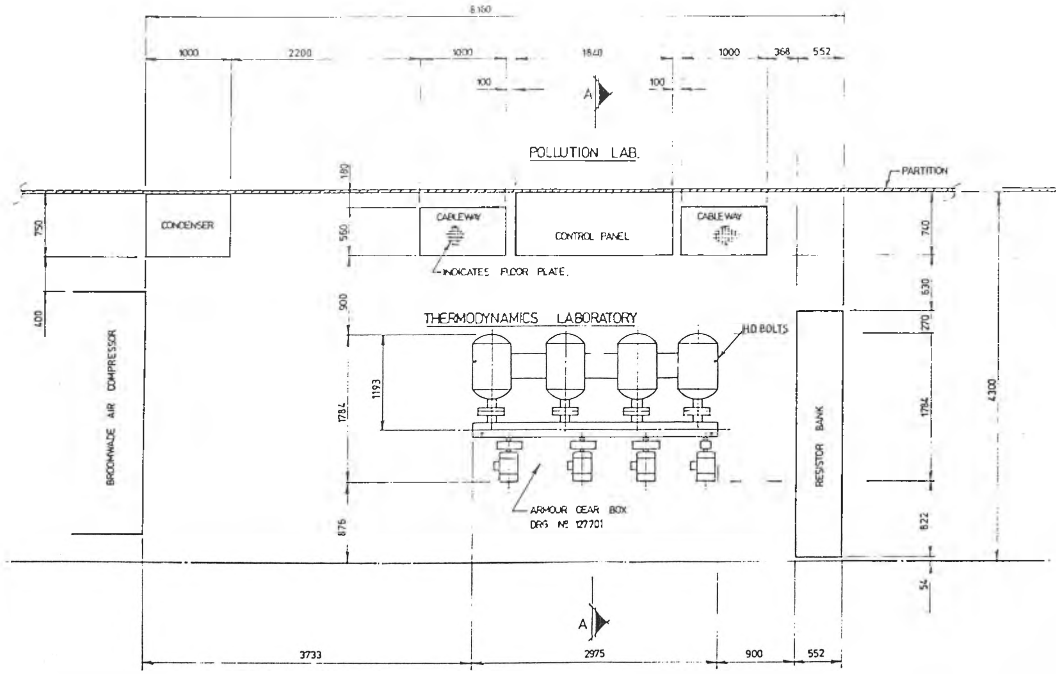
ALL WELDING PREPARATION & PROCEDURES NOT SHOWN SHALL CONFORM TO AS 1554

NO OFF	ITEM	DESCRIPTION		MAT L	REMARKS	CAT NO
2	M	GASKET	150×2×634	CORK		12113.134
2	L	GASKET	150×2×440	CORK		.133
4	K	SPACER		AS1204-250		.132
4	J	SPACER		AS1204-250		.131
2	H	BEARING HOUSING		AS1204-250		.130
2	G	BEARING HOUSING		AS1204-250		.119
2	F	BEARING HOUSING		AS1204-250		.128
2	E	BEARING HOUSING		AS1204-250		.117
2	D	BEARING HOUSING		AS1204-250		.116
2	C	BEARING HOUSING		AS1204-250		.115
2	B	BEARING HOUSING		AS1204-250		.114
2	A	BEARING HOUSING		AS1204-250		.113

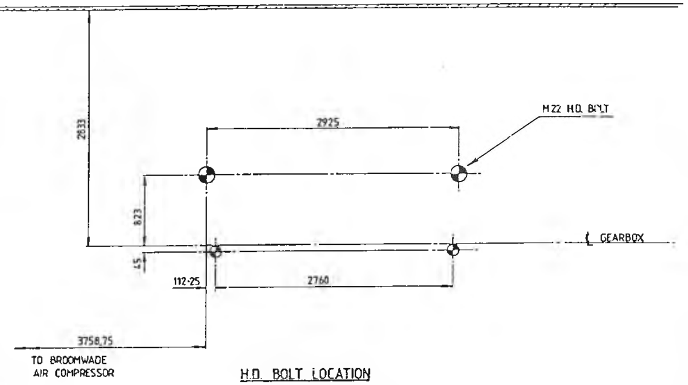
OPTIMUM DESIGN OF SPUR AND HELICAL GEARS. TO BS 436
THESIS GEARBOX TEST RIG
GEARBOX DETAILS SHT. 4 of 4
WOLLONGONG UNIVERSITY

LOCATION/JOB	DRN	CHKD	SCALE
REF DRG	TLH		1:1 & 1:2
DE	SDE	CDE	
DATE MAY 80			

AMDT 127709



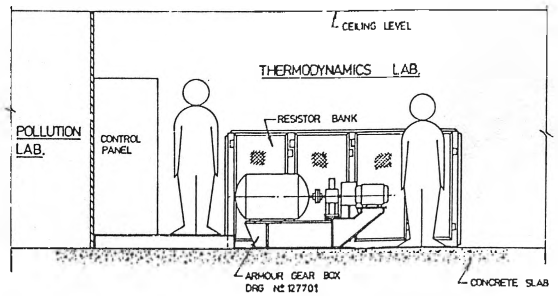
PLAN VIEW



H.D. BOLT LOCATION

REFERENCE DRAWINGS

127700	ARRANGEMENT & DETAILS RESISTOR BANKS
127701	ARRANGEMENT SH. 1
127702	ARRANGEMENT SH. 2
127703	GEAR BOX SUPPORT FRAME DETAILS
127704	FIRST MOTION SHAFTS
127705	MECHANICAL DETAILS
127706	GEAR BOX DETAILS SH 1
127707	GEAR BOX DETAILS SH. 2
127708	GEAR BOX DETAILS SH 3
127709	GEAR BOX DETAILS SH 4



SECTION A - A

NO OFF	ITEM	DESCRIPTION — MAT'L	REMARKS	CAT NO
		LOCATION/JOB: OPTIMUM DESIGN OF SPUR AND HELICAL GEARS TO BS 436 THESIS GEAR BOX TEST RIG LAYOUT OF EQUIPMENT IN LABORATORY		
		WOLLONGONG UNIVERSITY		

REF DRG	DRN D.G.	CHKD	SCALE 1 : 25
DE RJP	SJE	CDE	
DATE JULY '80		AMDT	127710

0 20 40 60 80 100 120 140 160 180 200 220 240 260 mm